THE MARINE STEAM TURBINE

2ND EDITION

J. W. SOTHERN, M.I.E.S.

THE MARINE STEAM TURBINE

(SECOND EDITION)

A Practical Description

OF THE

PARSONS MARINE TURBINE

AS PRESENTLY CONSTRUCTED, FITTED, AND RUN

INCLUDING

A Description of the Denny & Johnson Patent Torsion Meter for Measuring the transmitted Shaft Horse-Power, and containing Fifty Questions (with Solutions) on Elementary Turbine Design

BY

J. W. SOTHERN

Member, Institute of Engineers and Shipbuilders in Scotland; Hon. Member, West of Scotland Foremen Engineers' and Draughtsmen's Association; Principal, Sothern's Marine Engineering College, Glasgow; Member, Association of Engineering Teachers

Author of "Verbal Notes and Sketches for Marine Engineers," &c.

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AUTHOR'S PREFACE.

THE writer, having had exceptional opportunities of observing and studying the various details of marine turbine construction, has, in response to the growing demand for a work giving practical notes and sketches of this latest development of marine engineering, issued the present little work, and hopes it may fulfil to some extent the requirements of those anxious to obtain practical information regarding the Parsons marine steam turbine.

With few exceptions, the original sketches which illustrate the text were made by the author from the various working parts of actual turbines under construction, and can therefore be relied upon as being correct.

The author specially desires to thank the Council of the Institution of Engineers and Shipbuilders in Scotland for their kindness and courtesy in granting permission for the publication of copious extracts from the valuable paper on turbine construction by Mr E. M. Speakman, and read on 24th October 1905 at the forty-ninth meeting of that Institution.

The author has also to express his thanks to Messrs G. & J. Burns Limited for kind permission to reproduce the various photographs shown of the turbine machinery of their new Irish Channel Royal Mail Steamer "Viper," built and engined by the Fairfield Shipbuilding and Engineering Co. Limited, and intended to take up the daylight service runs between Ardrossan and Belfast.

The author also desires to tender thanks for the combined permission of Messrs Workman & Clark, Belfast, and Messrs D. Robertson & Co., Glasgow, to reproduce the photo of the turbines of the Allan Line steamship "Victorian," which forms the frontispiece of the book.

59 BRIDGE STREET, GLASGOW,
April 1906.

PREFACE TO SECOND EDITION.

THE marine steam turbine, being now fully recognised as a successful rival to the reciprocating engine, is receiving more respect at the hands of marine engineers, who are—somewhat reluctantly, it must be allowed—beginning to admit the good points of the former over the latter, notwithstanding previous condemnation of the turbine. The writer is, in fact, confident that in say fifteen years' time from now, the reciprocating engine will have become practically obsolete for all average speed and high speed steamers, and will be superseded by the turbine, which is, without any doubt or question, the ideal type of engine for marine work in particular.

In this edition of "The Marine Steam Turbine" more complete details relating to the practical construction and running of turbines are supplied than previously, and a large number of new original sketches and photographs further illustrate the text. The author specially desires to thank Professor Andrew Jamieson, M.I.C.E., &c., for very kindly correcting and revising part of the proofs of Section I. Readers can, as in the first edition, rely on the accuracy of both text and illustrations.

J. W. SOTHERN.

59 BRIDGE STREET, GLASGOW,
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SECTION I.

DEFINITIONS AND GENERAL PRINCIPLES.

BEFORE commencing to describe the chief features of the Parsons turbine, it is perhaps necessary to explain clearly the meaning of certain definitions which are closely connected with the theory and practice of this type as with all other types of steam engine.

Foot-Pound.—A foot-pound is the work done in raising a weight of one pound up through a distance of one foot.

Torque.—Torque is the turning movement to which a shaft is subjected when a force is exerted to rotate the shaft against a resistance such as that of the screw propeller in water. In ordinary engines the turning effort or torque is applied by means of the crank, and in turbines by the direct energy of the steam acting on the periphery of the blade circle of the rotor.

By arranging the rotor diameter so that the peripheral velocity of the blades is equal to about half that of the steam, the maximum amount of work in foot-pounds is extracted from each pound of steam passing through the turbine casing.

Heat.—Heat is merely a form of energy, and as such exists in two states—(1) in that of Potential or stored-up energy, and (2) in that of Kinetic or active energy. When

the molecules of a body or gas are set in rapid motion or vibration, heat is developed and work done. Consequently in the case of a steam engine, either of the reciprocating type or turbine type, the energy which produces rotation of the shaft is obtained by means of the transformation or heat energy into mechanical work.

British Thermal Unit (B.Th.U.).—This is taken as being equal to 778 foot-pounds of work or energy, and signifies that one heat unit, when transformed into mechanical energy, gives out 778 foot-pounds of work.

Saturated Steam.—Steam taken direct from the boilers is known as "saturated steam," as the density, or weight of water per cubic foot, is constant for any given pressure, as also is the temperature and volume. The steam supplied to all marine engines (without superheaters) is therefore of this quality, and calculations as to expansion, work done, fall of pressure, are usually made on this assumption. The steam supplied to the H.P. turbine of a turbine engine is therefore saturated steam. Sometimes the term "dry saturated steam" is used to distinguish this quality of steam from wet steam, or steam containing water from priming.

"Wet" Steam.—If water is carried off with the steam due to priming taking place in the boilers, the steam contains more water per cubic foot than is natural to the "saturation" pressure, volume, and temperature, and it is then known as "wet steam," or, "wet saturated steam."

Superheated Steam.—If saturated steam from the boilers is passed through the tubes of a superheater, the water contained in the steam is evaporated out of it, with the following results:—

1. Rise of temperature.
2. Increase of volume if pressure is kept constant; or,
3. Increase of pressure if volume is kept constant.

Properties of Saturated Steam.

Of from 0.5 *lb. to* 220 *lbs. Absolute Pressure per Square Inch.*

Absolute Pressure per square inch.	Temperatures.	Total Heat of One Pound of Steam from Water supplied at 32° F.	Total Latent Heat of Steam.	Volume of One Pound of Steam.
lbs.	Fahrenheit.	units.	units.	cubic feet.
0.5	80.2°	1105.5	1058.4	726.608
5	162.3	1130.9	1000.3	72.991
10	193.3	1140.3	978.4	37.845
15	213.1	1146.4	964.3	25.843
20	228.0	1150.9	953.8	19.710
25	240.1	1154.6	945.3	15.977
30	250.4	1157.8	937.9	13.459
35	259.3	1160.5	931.6	11.640
40	267.3	1162.9	926.0	10.267
45	274.4	1165.1	920.9	9.191
50	281.0	1167.1	916.3	8.322
55	287.1	1169.0	912.0	7.610
60	292.7	1170.7	908.0	7.011
65	298.0	1172.3	904.2	6.502
70	302.9	1173.8	900.8	6.059
75	307.5	1175.2	897.5	5.683
80	312.0	1176.5	894.3	5.348
85	316.1	1177.9	891.4	5.052
90	320.2	1179.1	880.5	4.790
95	324.1	1180.3	885.8	4.549
100	327.9	1181.4	883.1	4.335
105	331.3	1182.4	880.7	4.140
110	334.6	1183.5	878.3	3.963
115	338.0	1184.5	875.9	3.801
120	341.1	1185.4	873.7	3.652
125	344.2	1186.4	871.5	3.514
130	347.2	1187.3	869.4	3.388
135	350.1	1188.2	867.4	3.268
140	352.9	1189.0	865.4	3.159
145	355.6	1189.9	863.5	3.056
150	358.3	1190.7	861.5	2.960
155	361.1	1191.5	859.6	2.870
160	363.6	1192.3	857.8	2.785
165	366.0	1192.9	856.2	2.706
170	368.2	1193.7	854.5	2.631
175	370.8	1194.4	852.9	2.559
180	372.9	1195.1	851.3	2.493
185	375.3	1195.8	849.6	2.430
190	377.5	1196.5	848.0	2.370
195	379.7	1197.2	846.5	2.313
200	381.7	1197.8	845.0	2.263
210	386.0	1199.1	841.9	2.157
220	389.9	1200.3	839.2	2.065

The increase of latent heat with fall of pressure should be carefully noted, as the high efficiency of the low-pressure turbines is dependent on this.

The chief advantage of superheated steam lies in the fact that cylinder condensation is practically eliminated, as the steam does not then readily condense when exposed to cooled surfaces.

Another point of importance is that the specific heat of this steam being only .48, one B.T.U. of heat supplied to the steam has the effect of raising its temperature fully two degrees, as $1 \div .48 = 2.08$.

Adiabatic Expansion.—If steam expands in a cylinder or turbine casing, and neither receives heat from any external source nor gives out any heat externally, then the expansion is said to be "adiabatic," and all work done in the cylinder or turbine is obtained at the expense of the internal heat of the steam, which in falling in pressure and temperature conforms to this condition, and part of which condenses. In the cylinders of a marine engine of the reciprocating type, the expansion is approximately hyperbolic or isothermal, and in a turbine the expansion is approximately "adiabatic."

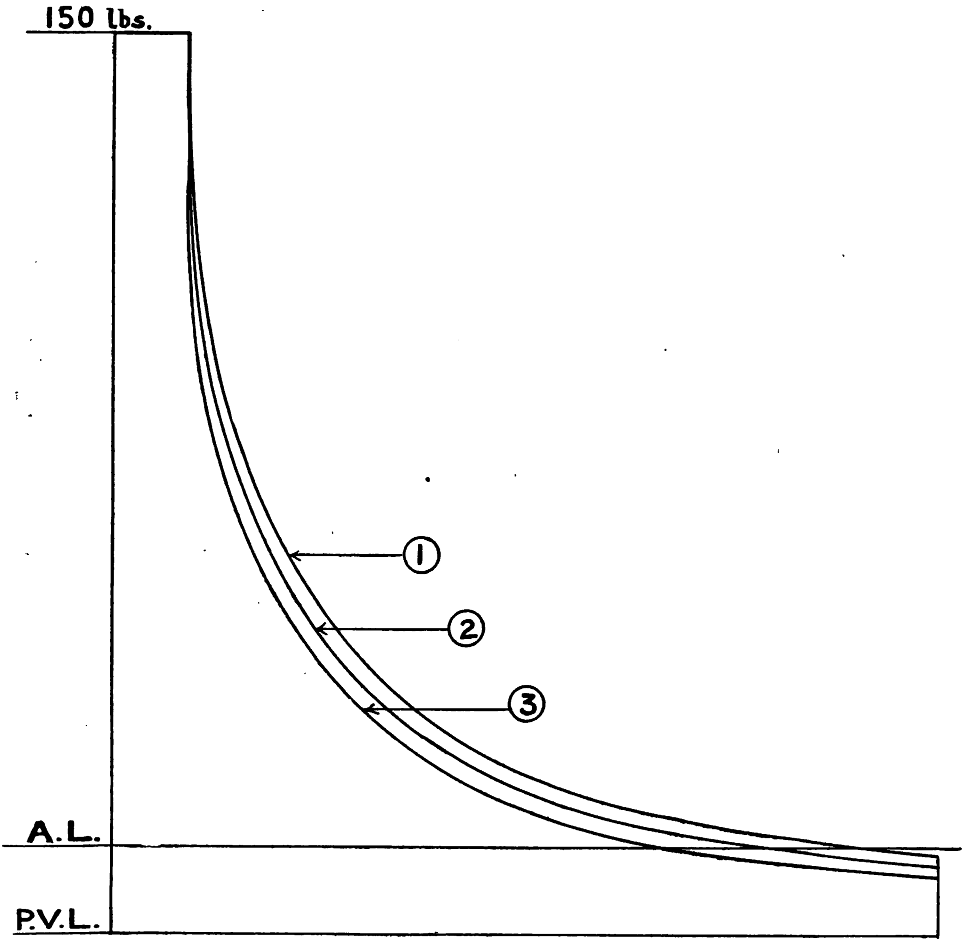
Hyperbolic or Isothermal Expansion.—This is founded on the well-known law of Boyle and Marriot that the pressure of a gas varies inversely as the volume; or, as it is expressed—

$$P_1 \times v_1 = p_2 \times V_2 = \text{Constant.}$$

Where P_1 = Initial pressure.
v_1 = ,, volume.
p_2 = Final pressure.
V_2 = ,, volume.

As explained before, ordinary steam, being an imperfect gas, does not exactly follow out the above law, but deviates in the direction of the "saturated steam" curve, as shown in the diagrams, in the case of reciprocating engine cylinders.

Dryness Fraction (or Factor).—In considering the actual work done by steam between the rows of blades of a turbine, it is important that the dryness fraction be



Expansion Curves of Steam.

(1) Isothermal or Hyperbolic Curve, $P \times V = \text{Constant}$ (Perfect Gas).

(2) Saturation Curve, $P \times V^{\frac{17}{16}} = \text{Constant}$ (Reciprocating Engine, approximately).

(3) Adiabatic Curve, $P \times V^{\frac{10}{9}} = \text{Constant}$ (Turbine Engine, approximately).

taken into account, as the result greatly depends on this quantity. After work is done by adiabatic expansion, the steam contains a certain amount of water, which proportionally reduces the internal heat still left in the steam. The dryness fraction is the ratio between the weight of dry steam per pound and the weight of the dry steam and water added together;

$$\text{Or, } \frac{\text{Weight of dry steam}}{\text{Weight of dry steam} + \text{weight of water}} = \text{Dryness Fraction.}$$

Suppose the water to be 25 per cent. of each pound weight,

$$\text{Then, } \frac{100-25}{100} = \frac{75}{100} = \frac{15}{20} = \tfrac{3}{4} = \text{Dryness Fraction (or Factor).}$$

So that after expansion and work done by the steam the actual units or foot-pounds of energy left are equal to the internal heat units multiplied by the fraction $\frac{3}{4}$.

Total Heat of Steam.—By the total heat of saturated, or boiler steam, is meant the number of heat units required to produce one pound of steam from a temperature of 32° Fahr. to any given temperature and pressure. The total heat includes the latent heat of steam formation and the sensible or thermometer heat.

RULE.—$1083 + .3 \times T° =$ Total Heat (above 32° Fahr.).

Where $T°$ = Temperature of the steam (Fahr.).

Internal Heat of Steam.—By this is meant the heat or energy required to change one pound of water into steam at any given pressure.

External Heat of Steam.—By this is meant the heat required to produce increase of volume (water to steam) against an external resistance or pressure.

Latent Heat of Steam.—The sum of the Internal heat and External heat is equal to the latent heat.

The Latent Heat can be calculated as follows:—

RULE.—$1114 - .7 \times T° =$ Latent Heat.

Where $T°$ = Temperature of the steam (Fahr.).

EXAMPLE.—Calculate the Total Heat, Latent Heat, and Sensible Heat of 1 lb. of steam at 160 lbs. pressure by gauge.

$160 + 15 = 175$ lbs. gross pressure and $371°$ Temperature (nearly).

Then, $1083 + .3 \times 371 = 1194.3$ Total Heat,

and $1114 - .7 \times 371 = 854.3$ Latent Heat.

Therefore $371° - 32° = 333.9$ Sensible Heat.

NOTE.—The above are all calculated from a temperature of 32° Fahr.

Potential Energy is the energy contained or stored up in steam of a given pressure and temperature, the amount of energy contained increasing with the pressure and the temperature.

Kinetic Energy is the result of setting free the potential or stored-up energy of the steam, which then shows as active energy in the performance of work. In a steam engine the steam acts on the pistons, and by causing motion to take place work is done, and, as a result, the steam falls in pressure and in temperature. In a turbine, the steam at a certain pressure and velocity leaves the first row of guide blades, and striking the first row of moving blades gives up part of its kinetic energy, which results in a decrease in pressure and in heat. It then enters the next row of guide and moving blades, where more energy is given up, and a further decrease in pressure and in heat takes place. This is repeated row after row, the steam falling in pressure and in temperature, but, be it noted, increasing in volume. This increase of volume would produce increase of velocity if the blades were not made (1) longer, or (2) spaced farther apart. Both methods, separately and combined, are adopted at different expansion stages of the Parsons turbine, as will be shown later.

The effective kinetic energy of each pound of steam supplied to the turbine is employed in exerting a torsional stress on the shaft, and thus produces rotation. In the boilers the potential energy of the steam is generated, and in the cylinders of an ordinary engine, or in the turbine casing of a turbine engine, the potential energy is liberated

and transformed into kinetic energy, and a certain number of foot-pounds of work are done in causing rotation of the shaft. In all steam engines only a very small portion of the total heat of the steam can be changed into foot-pounds of useful work, seldom more than about 14.3 per cent., or $\frac{1}{7}$, as shown by the following:—

Suppose the consumption of coal is 1.5 lb. per I.H.P. per hour—

$$\text{Then, } \frac{60 \times 33000}{1.5 \times 11500 \times 778} = \frac{1}{7}, \text{ or } 14.3 \text{ per cent.}$$

NOTE.—Allow 11500 units of heat per pound of coal.

This applies equally to a turbine or reciprocating engine. Suppose, then, that a pound of steam in passing through a turbine gives up, say, 200 units of heat,

Then, $200 \times 778 = 155600$ foot-pounds of energy given out.

This means that 155600 foot-pounds of work have been done in rotating the shaft, neglecting blade leakage, blade friction, and other losses.

If, then, we know that a certain number of pounds of steam are used in a given time, say, for example, 1000 lbs. per minute, then,

$1000 \times 155600 = 155600000$ foot-pounds of energy developed.

After losses are allowed for, the actual foot-pounds of work done, or kinetic energy expended, is exactly equal to the units of heat effectively applied, multiplied by 778.

Blade Friction.—It is important that the surfaces of the blades should be as smooth as possible, since the friction produced by the flow of steam across the blades seriously affects the efficiency of the turbine. It will also be obvious that water in the steam will produce a similar result, so that one clear advantage of superheating would be the reduction in frictional resistance set up by saturated steam.

Expansion of Steam.—In a modern marine triple-expansion engine with cylinder areas of say H.P. to L.P. as 1 is to 7.5, and with a cut-off in the H.P. of $\frac{1}{3}$ stroke, the total number of expansions of steam would be 22.5, as $7.5 \times 3 = 22.5$.

In the turbine engine, however, the number of expansions of steam is much more than this, from 125 to 140 expansions being readily obtained. With an H.P. turbine initial pressure of 150 lbs., and a condenser vacuum of 29 in., or back pressure of say 1 lb., the steam would expand about 150 times. From this it will be seen that more work can be got out of the steam by the greatly increased number of steam expansions, and the importance of obtaining a very high vacuum in the condensers will be obvious.

Condensation.—In all ordinary engine cylinders losses from condensation are more or less serious, the cause being, as is well known, the difference of temperature existing between the initial and exhaust pressures, alternately heating up and cooling down the cylinder walls. This results in a certain amount of steam condensing into water without doing work. To take an example:—

Suppose the H.P. initial pressure in a triple-expansion engine to be 160 lbs. gauge, and the M.P. receiver pressure 54 lbs. gauge,

$$\text{Then, } 160 \text{ lbs. gauge} = 370^\circ \text{ Temp.,}$$
$$\text{and } 54 \text{ lbs. gauge} = 300^\circ \text{ Temp.}$$

Now during admission the temperature of the steam is 370°, and during exhaust about 300°,

$$\text{Therefore } 370^\circ - 300^\circ = 70^\circ \text{ drop of Temp.}$$

This drop of heat produces, as stated above, a serious condensation loss, as the hot initial steam is condensed for a certain period following admission, owing to the cooling down of the cylinder walls during exhaust.

In a turbine casing no such variation in temperature exists, as the range of temperature is practically constant throughout the turbine from end to end, the steam entering at one end at a high temperature, and flowing to the other end continuously as it falls gradually in pressure and temperature.

Again, if condensation does take place, the water formed is not so troublesome to get rid of as in an ordinary cylinder, as it simply drains away to the exhaust end of the turbine and so to the condenser, or air pump.

The condensation which does occur in the turbines is that due to the adiabatic expansion of the steam during the transformation from potential to kinetic energy, as mentioned elsewhere.

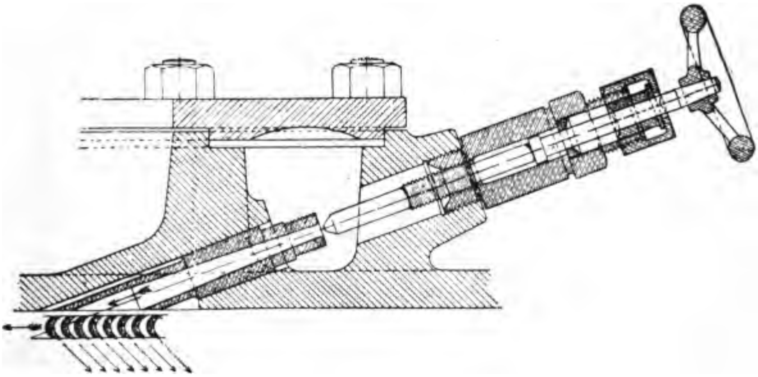
Principle of Turbine.—The steam turbine is a machine designed to convert the kinetic energy of steam into direct rotary motion. The two principal types of turbine are—(1) Impulse Turbines, those arranged with an expanding nozzle in which the high velocity of discharge impinges against a series of small buckets secured on the circumference of a large wheel keyed to the driving shaft, the De-Laval turbine being an example of this type; and those (2) Impulse-reaction Turbines, in which the steam passes through a number of rings of fixed blades and of moving blades, expanding as it travels, an example of which is found in the Parsons turbine.

Work by impulse is produced by high velocities, and as the work is done at the expense of the internal heat, water is formed, which thus diminishes the heat left.

The Parsons turbine is generally called a reaction turbine, although the correct term should be "impulse-reaction" turbine, as the steam actually does act first by impulse from the guide to the moving blades and afterwards works by reaction from the moving to the guide blades.

De-Laval Turbine.—In the specially shaped diverging nozzle of the De-Laval turbine shown in the sketch, the steam expands down to the required exhaust pressure, and the resultant kinetic energy acquired is applied direct to the small buckets or vanes, the steam being in consequence at a very high velocity. To obtain the best efficiency the circumferential velocity of the turbine blades should be equal to about half the velocity of the steam, and this, of course, demands a very high revolution speed. In the De-Laval turbine the speed is often as high as 20,000 revolutions per minute; this can, however, be reduced by suitable gearing to about 2,000 revolutions per minute, but as even this is too high for the shafting of

marine engines, the non-adaptability of this turbine for marine purposes will be obvious. The steam is admitted to the nozzles (usually four or six in number) and controlled by regulating hand-valves.



ARRANGEMENT OF NOZZLE AND SHUTTING-OFF VALVE.

De-Laval Turbine.

It is worthy of notice that in this type of turbine the turbine wheel is rotated by steam at the expanded or lowest pressure, as the actual expansion takes place in the nozzle, which is specially designed for that purpose.

The De-Laval type of turbine is much in use for the driving of dynamos, and many steamers are supplied with this turbine for the lighting of the ship.

Parsons Turbine.—In this, the latest and most successful development of marine engineering, steam is admitted direct from the boilers to vanes or blades on the shaft, thus doing away with the necessity for piston valves or slide valves, cylinders, pistons, piston rods, crossheads, connecting rods, cranks, eccentrics, eccentric rods, and links, &c.

The power to rotate the shaft is therefore applied direct, and this in itself constitutes one of the conditions of an ideal engine. The inventor, the Hon. C. A. Parsons, M.A., F.R.S., gives the following brief description of the turbine :—

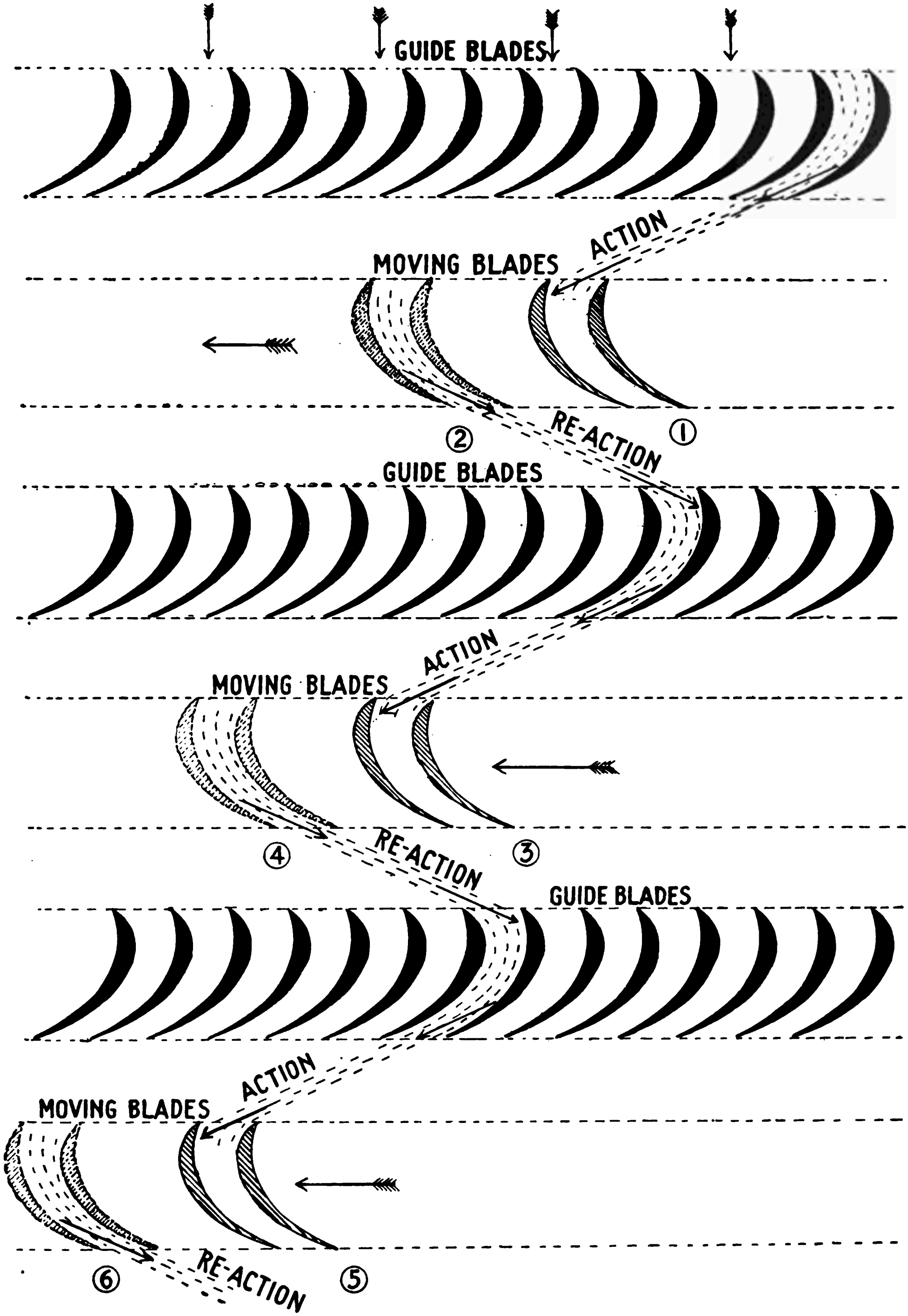
" The Parsons turbine consists of a cylindrical case with numerous rings of inwardly projecting blades. Within

this cylinder, which is of variable internal diameter, is a shaft or spindle, and on this spindle are mounted blades, projecting outwardly, by means of which the shaft is rotated. The former are called fixed or guide blades, and the latter revolving or moving blades. The diameter of the spindle is less than the internal diameter of the cylinder, and thus an annular space is left between the two. This space is occupied by the blades, and it is through these the steam flows. The steam enters the cylinder by means of an annular port at the forward end; it meets a ring of fixed guide blades which deflects it so that it strikes the adjoining ring of moving blades at such an angle that it exerts on them a rotary impulse. When the steam leaves these blades it has naturally been deflected. The second ring of fixed blades is therefore interposed, and these direct the steam on to the second ring of rotating blades. The same thing occurs with succeeding rings of guide and moving blades until the steam escapes at the exhaust passage."

Steam from the boiler is admitted by suitable hand valves to the forward end of the casing surrounding the blades, and after passing through a ring of guide blades fixed to the casing, strikes the first ring of shaft or rotor blades; it next passes through the second ring of fixed blades, then the second ring of rotor blades, and so on, passing alternately ring after ring of guide and rotor blades, and so rotating the shaft, until it finally exhausts at the other end of the casing at a reduced pressure.

Parallel Flow.—Parsons' marine turbine is known as that of the impulse and reaction "parallel flow" type, as the steam enters the guide vanes in lines parallel more or less to the shaft axis, and in this way passes from end to end of the turbine, reacting, expanding, and falling in pressure as it travels.

Action of Steam.—As will be seen from the foregoing, the steam striking the blades imparts a turning movement to the shaft, and after reacting and passing through the series of rings of vanes of the H.P. turbine



Path traced out by Steam in Parsons' Turbine.

exhausts simultaneously into the two L.P. turbines, one on either side, and expanding through the longer casing and shaft blades of these turbines, finally exhausts, at a low absolute pressure, from 1½ to 3 lbs., into the condensers, one for each L.P. turbine.

Flow of Steam through Blades.—The diagram on page 13 shows graphically the path followed out by the steam as it passes through each successive ring of fixed and moving blades. Observe that the steam, after passing through the first ring of guide blades, strikes the first ring of rotor or moving blades and by the action set up assists in rotating the shaft; by the time the steam has changed its direction the rotor has moved round a certain distance (from 1 to 2), and the reaction of the steam, due to its somewhat sudden change of direction, still further assists in rotating the shaft. The steam then leaves the rotor blades and enters the next ring of guide blades, where, after again being deflected in its path, it enters the next ring of moving blades, where the action and reaction process is again repeated; leaving the second ring of moving blades at position 4 the steam enters the third ring of guide blades at a point 5 still farther round the circumference, and so on for each of the following rings. The steam thus describes a somewhat zigzag path in passing along the rotor, its direction being not unlike that of a screw thread. Work is done at each ring of blades and heat given up, expansion of the steam taking place in due proportion, so that the velocity of flow increases, and to allow for this the lengths and spacings of the blades must be increased to maintain the same ratio between the blade velocity and the steam velocity, upon which the turbine efficiency depends.

The diagram shows the imaginary path described by a small portion of steam, and the dotted blades show the circumferential advance of the rotor blades at each ring, which produces the thread-like path traced out by the steam.

Increase of Steam Volume.—To allow of the steam increasing in volume, as fall of pressure takes place,

the various sets of blades increase in length from the forward to the after end, the clearance spaces between the blades also increasing in proportion, which necessitates packing pieces of a larger size being employed. The blades also vary in shape or curvature, being flatter in section aft than forward. Each set of blades for each expansion requires its own allowance for expansion of metals by heat, so that the working clearance between the blades and casings or drums increases slightly throughout the turbine from forward aft. One of the practical difficulties met with in turbine construction at present is the correct adjustment for this expansion, as slight mishaps have occurred in one or two instances owing to fouling of the parts when heated up, the clearance allowance being insufficient.

Strictly, each successive ring of blades should be either of a wider pitch or greater height than the preceding one, as the steam is continuously falling in pressure and expanding in volume, but the Hon. C. A. Parsons considers the present arrangement quite near enough for practical purposes for all the difference that results.

Theoretical Blade Heights.—The next diagram shows how the blade heights would vary if made to exactly correspond to the steam expansion.

As the pressure falls the volume increases (adiabatically), so that each succeeding row of blades should be slightly longer than the preceding row. In practice, however, this is not carried out, the blades being arranged in sets of equal height, or, as it is called, "stepped." The horizontal dotted lines show the actual arrangement of blade heights as usually fitted. The 6th, 7th, and 8th expansions of each L.P. turbine consist of rows of blades of equal height, but it should be noted that the angle or curvature of each set is different, the 7th and 8th expansions having blades of flatter section and of wider circumferential pitch than those of the preceding sets.

It will be observed that the first expansion of the L.P. has *shorter* blades than the last expansion of the H.P. It must, however, be remembered that the diameter of the

L.P. drums is *more* than the H.P., thus giving a higher peripheral and steam speed, also that only half the quantity of steam passes through each L.P. turbine.

NOTE.—Observe that the total number of blades is the same for the H.P. turbine and each L.P. turbine, the H.P., however, only having four expansions, each one containing *twice* the number of blades contained in each of the eight expansions of the L.P. turbines.

Velocity Calculations, &c.—The foot-pounds of energy contained in a given weight of steam at a given pressure and velocity are found as follows:—

$$\text{Kinetic Energy} = \frac{W \times V^2}{2\,g} \text{ foot-pounds.}$$

NOTE.—W = Weight of steam in pounds.

V = Velocity of steam in feet per second.

g = gravity's acceleration, or 32.2 feet per second, per second.

The change in kinetic energy or the work done between the blades may be expressed as follows, friction and other losses, such as tip clearance leakage, being neglected.

$$\text{Kinetic Energy} = \frac{(V^2 - v^2) \times W}{2\,g} \text{ foot-pounds.}$$

Where V = Velocity of steam in feet per second at the entering edge of blades.

v = Velocity of steam in feet per second at the leaving edge of blades.

W = Weight of steam in pounds.

g = gravity's acceleration, or 32.2 feet per second, per second.

For example—Let V = 600 feet per second.

„ W = 1 pound.

Then, $\frac{W \times V^2}{64.4} = \frac{1 \times 600^2}{64.4} = 5590$ foot-pounds of kinetic energy.

778

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For example—Let $V = 400$ feet per second.

" $v = 300$ " "

" $W = 1$ pound.

Then, $\frac{(V^2 - v^2) \times W}{64.4} = \frac{(400^2 - 300^2) \times 1}{64.4} = 1087$ foot-pounds in passing across one row of blades.

NOTE.—In turbine practice the steam expands approximately adiabatically. No heat being supplied from without, and if no leakage of heat takes place, the work is done at the expense of the internal heat energy of the steam, and the fall in pressure, in temperature, and amount of condensation in the turbine is proportional to the work done.

The expansion being adiabatic, the heat drop for a given pressure drop is much more than that shown in the total heat table of saturated steam, as the steam which condenses during the performance of work reduces the weight of actual steam remaining after expansion, and therefore the heat contained per pound of steam and water mixture is proportionally less. It should be again noted that the internal heat energy of the steam is transformed into mechanical energy, hence the heat drop so often referred to.

Let V = Velocity of steam.

" W = Weight.

" H = Heat units given up.

$64.4 = 32.2 \times 2$ (gravity's acceleration per sec., per sec.).

Then, $\frac{V^2 \times W}{64.4}$ = Foot-pounds of kinetic energy.

And, Foot-pounds $\times 64.4 = V^2 \times W$.

Therefore, $H \times 778 \times 64.4 = V^2 \times W$.

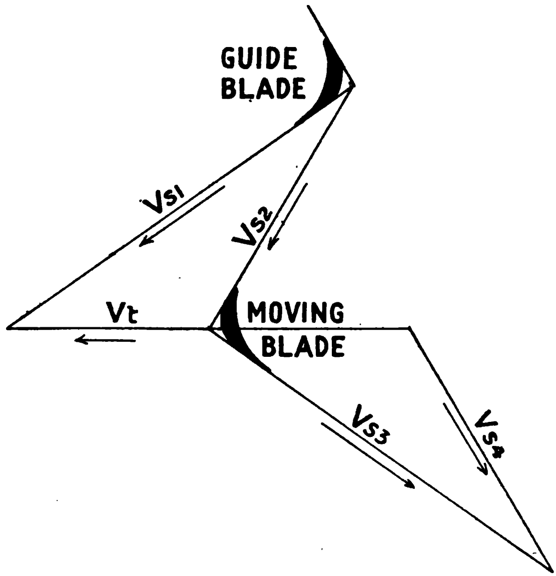
" $V^2 = \frac{H \times 778 \times 64.4}{W}$

" $V = \sqrt{\frac{H \times 778 \times 64.4}{W}}$

" $\frac{V^2 \times W}{64.4} = H \times 778$.

And, $\frac{\text{foot-pounds}}{778} = H$.

Velocity Diagram.



The above diagram shows graphically the varying steam velocities previously referred to.

V_{S1} = absolute steam velocity (initial).
V_{S2} = relative „ „ „ with regard to blade or rotor speed.
V_{S3} = relative „ „ leaving the blades.
V_{S4} = absolute „ „ „ „
V_t = blade velocity.

The actual velocity of the steam passing between the rows of blades depends upon the drop of pressure produced by the expenditure of foot-pounds of heat energy developed as work done.

The velocity of the steam in feet per second, due to any pressure drop, is found as follows :—

$$\text{Velocity of steam} = \sqrt{64.4 \times 778 \times (H_1 - H_2)}.$$

NOTE.—H_1 = Initial heat units.
H_2 = Final „ „
64.4 = twice 32.2 (gravity acceleration per sec., per sec.).

EXAMPLE.—Calculate the velocity of the steam between the blades of a Parsons turbine if the heat units per pound of steam at admission edge are 1,200, and at exhaust edge 1,178 units (a heat drop of 2 units, neglecting frictional and other losses).

$$\text{Then, } \sqrt{64.4 \times 778 \times (H_1 - H_2)} = \sqrt{64.4 \times 778 \times (1200 - 1178)}$$
$$= 316 \text{ ft. Velocity per second.}$$

EXAMPLE.—Calculate the energy contained in one pound of the above steam.

RULE.—$\frac{V^2 \times W}{64.4}$ = foot-pounds; Therefore, $\frac{316^2 \times 1}{64.4} = 1556$ foot-pounds, and $1556 \div 778 = 2$ B.T. Units as before.

Absolute Velocity and Relative Velocity.—By absolute velocity is meant the velocity of the steam with regard to stationary objects.

By relative velocity is meant the velocity of the steam with regard to moving objects, in this case with regard to the blade velocity.

The two steam velocities referred to are shown graphically in the sketch, and it will be observed that the rotor velocity affects the relative velocity of the steam inversely, that is, with high rotor speed the relative steam speed on leaving the moving vanes is low, and *vice versâ*.

Guide Blades.—In the guide blades, the work done is equal to the following:—

$$\frac{V_{S1}^2 - V_{S4}^2}{64.4} = \text{foot-pounds in guide blades.}$$

Moving Blades.—In any of the moving blades, the work done is equal to the following:—

$$\frac{V_{S3}^2 - V_{S2}^2}{64.4} = \text{foot-pounds in moving blades.}$$

Notice that the work done in the moving blades is calculated from the *relative* steam velocity of admission and *relative* steam velocity of exit, whereas the work done

in the guide blades is calculated from the absolute steam velocities of both admission and of exit.

Work of Steam Acceleration.—The work done by the steam upon itself in the way of increased velocity in the first row of stationary or guide blades of a turbine is, as explained above, found as follows :—

$$\frac{V^2 \times W}{64.4} = \text{foot-pounds.}$$

EXAMPLE.—Calculate the foot-pounds of energy developed by one pound of steam within the guide blades of turbine if the steam velocity is 300 feet per second.

$$\text{Then, } \frac{V^2 \times W}{64.4} = \frac{300^2 \times 1}{64.4} = 1400 \text{ foot-pounds (nearly).}$$

In any turbine, then, the total energy per pound of steam is equal in foot-pounds to the following :—

$$(1114 + .3 \times T^\circ) \times 778 = \text{foot-pounds contained.}$$

Therefore, after the steam has passed through a pair of rows of blades, the equation will be as follows :—

$$(1114 + .3 \times T^\circ) \times 778 = \frac{V^2}{64.4} + H \times 778.$$

Where H = heat left in the steam after fall in pressure between rows.

From the above it will be obvious that the total energy remains the same, and is exactly equal to the work done added to the energy still remaining in the steam, part of which will be represented by the small per cent. of water condensed, and the remainder by the actual heat velocity.

EXAMPLE.—At the inlet edge of the blades the pressure is 140 lbs., and at the outlet edge 138 lbs., the difference of B.T. units due to the nature of the expansion being 2.5 $(H_1 - H_2)$. Calculate (1) the steam velocity, (2) the kinetic energy of the steam.

$$V = \sqrt{64.4 \times (H_1 - H_2) \times 778}.$$

$$\text{Therefore, } \sqrt{64.4 \times 2.5 \times 778} = 354 \text{ feet per sec.}$$

$$\text{and, } \frac{V^2}{64.4} = \text{kinetic energy} = \frac{354^2}{64.4} = 1945 \text{ foot-pounds.}$$

EXAMPLE.—The two velocities of the steam are $V_1 = 300$ feet per second, and $V_2 = 450$ feet per second. Calculate the foot-pounds of work given up, and the number of B.T. units of heat drop across the blades per pound of steam.

$$\text{Kinetic energy} = \frac{(V_2^2 - V_1^2) \times W}{64.4} = \frac{(450^2 - 300^2) \times 1}{64.4} = 1746.8 \text{ ft. lbs.}$$

$$\text{Heat drop} = \left(\frac{V^2 \times W}{64.4}\right) \div 778 = \left(\frac{450^2 \times 1}{64.4}\right) \div 778 = 4.04 \text{ units of heat.}$$

Notice that the relative velocity of the entering steam is less than its absolute velocity, and that the relative velocity of the steam leaving the moving blades is *more* than its absolute velocity. This is due to the velocity of the blades being in one constant direction.

Blade Velocity.—The mean velocity of the blades being often from .4 to .5 of the steam velocity, then the steam velocity × .5 = blade velocity at mean diameter of the blade circle. So that if the steam velocity is, as stated, 18000 feet per minute, then,

$$18000 \times .5 = 9000 \text{ feet per minute.}$$

Now if the revolutions are to be not more than, say, 400 per minute, the diameter of rotor can be found as follows :—

$$\frac{9000}{400 \times 3.1416} = 7.08 \text{ feet diameter across mean diameter of blades.}$$

The actual diameter of rotor would probably be somewhere under 7 feet, when the length of blades is deducted.

Rotor Diameter.—The great difference in turbine diameter required for varying revolutions is shown by the following cases :—

Let V_s = steam velocity, and V_t = blade velocity.

EXAMPLE.—Steam velocity $V_s = 21000$ feet per minute, and if $\frac{V_t}{V_s} = .5$, find the diameter of rotor required for (1) 600 revolutions, and (2) 200 revolutions per minute.

Then (1) $21000 \times .5 = D \times 3.1416 \times 600$.

$$\therefore \frac{21000 \times .5}{3.1416 \times 600} = 5.5 \text{ feet diameter of rotor.}$$

Again, (2) $21000 \times .5 = D \times 3.1416 \times 200$.

$$\therefore \frac{21000 \times .5}{3.1416 \times 200} = 16.7 \text{ feet diameter of rotor.}$$

NOTE.—The above diameters are measured across the mean *height* of blades, so that the actual diameter of rotor might be, say, 9 or 10 inches less than the above results.

Increase of Velocities.—At each succeeding expansion, the mean diameter of the blades being increased owing to increase of blade height, the velocity of the blades per second at the after or exhaust end of any turbine rotor is rather more than at the admission or forward end. An example will make this clear.

EXAMPLE.—Calculate the respective blade velocities at each expansion of the H.P. turbine, if the blade heights are as follows :—

$1\frac{1}{16}$ in. at 1st expansion.
$1\frac{3}{8}$ in. " 2nd "
$1\frac{7}{8}$ in. " 3rd "
$2\frac{1}{2}$ in. " 4th "

The diameter of rotor is 3 feet 6 inches, and the revolutions per minute 600. Diameter of rotor = 3 feet 6 inches = 42 inches, then

At 1st expansion, 42 in. + $1\frac{1}{16}$ in. = $43\frac{1}{16}$ in. = 43.0625 mean diameter across blades.

At 2nd expansion, 42 in. + $1\frac{3}{8}$ in. = $43\frac{3}{8}$ in. = 43.375 mean diameter across blades.

At 3rd expansion, 42 in. + $1\frac{7}{8}$ in. = $43\frac{7}{8}$ in. = 43.875 mean diameter across blades.

At 4th expansion, 42 in. + $2\frac{1}{2}$ in. = $44\frac{1}{2}$ in. = 44.5 mean diameter across blades.

Therefore, $\frac{43.0625 \times 3.1416 \times 600}{60 \times 12} = 112$ ft. per second velocity of blades at 1st expansion.

Therefore, $\frac{43.375 \times 3.1416 \times 600}{60 \times 12} = 113$ ft. per second velocity of blades at 2nd expansion.

Therefore, $\frac{43.875 \times 3.1416 \times 600}{60 \times 12} = 114$ ft. per second velocity of blades at 3rd expansion.

Therefore, $\frac{44.5 \times 3.1416 \times 600}{60 \times 12} = 116$ ft. per second velocity of blades at 4th expansion.

NOTE.—Divide by 60 seconds and by 12 inches to obtain feet per second.

NOTE.—The blade tip clearance, about $\frac{50}{1000}$ of an inch, or .05 in., is neglected in the above calculations.

It should be noted that the steam velocities increase correspondingly at each successive expansion, so that the average ratio of blade velocity to steam velocity, or $\frac{V_t}{V_s}$, is still maintained throughout the turbine.

From the foregoing, it will be easily seen that the blade heights and openings must increase at a high ratio to allow for the very rapid increase of volume at the lower pressures carried near the exhaust end of the low-pressure turbines.

Steam Velocity.—The velocity or speed of steam at any given pressure varies according to the pressure of the exhaust or opposing back pressure. In marine turbine practice, the velocity of the steam is often about 300 feet per second, or 18,000 feet per minute.

For example, at an initial pressure of 180 lbs. the velocity of steam when flowing directly into the atmosphere is equal to about 3,000 feet per second, and if allowed to flow into a vacuum of 25 inches the velocity is about 3,700 feet per second.

After passing each row of rotor blades a small drop in pressure and in heat energy takes place owing to the useful work done by the steam in rotating the shaft. This drop is proportional to the foot-pounds of energy expended at each row of rotor blades, and, as before stated, the exchange of heat into work or kinetic energy is equal to 778 foot-pounds per unit of heat. The average drop in pressure per row of blades throughout a marine turbine works out as about .76 of a pound.

Heat Drops.—The fall of pressure, temperature, and heat units in the casing of a turbine may be described in a somewhat rudimentary form as follows:—As previously

explained, the necessary steam velocity required to produce effective work, or kinetic energy, on the blades of the turbine rotor is obtained by allowing a suitable drop in pressure and in heat units between the successive pairs of rows of blades. It is important to note that the average drop per pair of rotor blade rows is less in the L.P. turbines than in the H.P. turbine. This is accounted for by the fact that the difference in total heat units of steam at low pressure is more than that of high pressure for the same pressure drop, or in other words, an equal difference in drop of heat units can be obtained by a smaller drop of pressure. The difference in drop of heat units necessary for the required foot-pounds of energy to be developed can therefore be obtained by a smaller pressure drop per pair of rows with low-pressure steam, as the following examples will perhaps make clear :—

Work done during Adiabatic Expansion.—To calculate the work done, or, which is the same thing, the units of heat given up or converted into work during the adiabatic expansion of steam in a turbine, the following data are required :—

The absolute temperature of the steam before and after expansion.

The latent heat of the steam before and after expansion.

The dryness factor of the steam before and after expansion.

Let T_1° = Absolute temperature before expansion.
T_2° = „ „ after „
H_1 = Latent heat before expansion.
H_2 = „ „ after „
f_1 = Dryness factor before expansion.
f_2 = „ „ after „

The heat energy given out in British Thermal Units =

$$f_1 \times H_1 - f_2 \times H_2 + T_1^\circ - T_2^\circ = \text{B.T.U.}$$

And, $\text{B.T.U.} = \dfrac{V^2}{2g \times 778}$; or, $V^2 = 2g \times 778 \times \text{B.T.U.}$

Therefore, $V = \sqrt{2g \times 778 \times \text{B.T.U.}}$

NOTE.—V = Velocity of steam in feet per second.
$2g = 2 \times 32.2$ = Acceleration due to gravity in feet per second per second.

NOTE.—The "Entrophy" diagram affords the clearest explanation of the heat utilisation and expenditure.

EXAMPLE 1—**High-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 150 lbs. absolute, to another expansion where the pressure is 140 lbs. absolute, given that—

$$\begin{aligned} T_1^\circ &= (358^\circ + 461^\circ) = 819. \\ T_2^\circ &= (353^\circ + 461^\circ) = 814. \\ H_1 &= 861. \\ H_2 &= 865. \\ f_1 &= 1. \\ f_2 &= .996. \end{aligned}$$

NOTE.—Absolute temperature = Fahrenheit + 461°.

$$\begin{aligned} \text{Heat units given up} &= 1 \times 861 - .996 \times 865 + 819 - 814 \\ &= 861 - 861.54 + 819 - 814 = 1680 - 1675.5 = 4.5 \text{ B.T.U.} \end{aligned}$$

Also, calculate the velocity of the steam.

$$\text{Then, } V = \sqrt{64.4 \times 778 \times 4.5} = 474 \text{ feet per second.}$$

EXAMPLE 2—**Low-Pressure Steam.**—In a marine turbine find the heat energy given up by one pound of steam when flowing from one expansion where the pressure is 6 lbs. absolute, to another expansion where the pressure is 2 lbs. absolute, given that—

$$\begin{aligned} T_1^\circ &= (170^\circ + 461^\circ) = 631^\circ. \\ T_2^\circ &= (126^\circ + 461^\circ) = 587^\circ. \\ H_1 &= 995. \\ H_2 &= 1026. \\ f_1 &= .85. \\ f_2 &= .8. \end{aligned}$$

$$\begin{aligned} \text{Heat units given up} &= .85 \times 995 - .8 \times 1026 + 631 - 587 \\ &= 845.75 - 822.8 + 631 - 587 = 1476.75 - 1409.8 = 66.95 \text{ B.T.U.} \end{aligned}$$

Also, calculate the velocity of the steam.

$$\text{Then, } V = \sqrt{64.4 \times 778 \times 66.95} = 1831 \text{ feet per second.}$$

From the foregoing it will be seen that in Example No. 1, with a pressure drop of 10 lbs. (from 150 to 140 lbs.) and the conditions stated, the number of heat units converted into work, or kinetic energy, is only 4.5, whereas in

Example No. 2, with a *pressure drop of only* 4 *lbs.* (from 6 lbs. to 2 lbs. absolutely), the number of heat units converted into kinetic energy is 66.95, thus clearly indicating the greater value of lower pressure steam in turbine practice, and accounting for the two L.P. wing turbines each developing equal power to the centre H.P. turbine with about half the weight of steam at a very much lower pressure.

The B.T.U. or heat drop for a given pressure drop *increases with the fall of pressure*, and as the kinetic energy given up to the blades entirely depends on the *heat drop*, it naturally follows that in the case of very low-pressure steam the same amount of work can be obtained with a much smaller pressure drop, as also shown in the example from actual practice given on page 106. It should be carefully noted that in turbine practice the kinetic energy got out of the steam at each stage or pair of rows varies directly as the drop in heat units, and is, strange as it may appear, quite independent of the pressure carried. It will also be obvious that a high vacuum will increase the efficiency of the low-pressure turbines by allowing of a further drop in pressure, thus developing to the full extent the benefit of the increasing drop in heat units per given pressure drop. Allowing, therefore, equal powers to be developed in each shaft and in each turbine, it is of interest to note that—

1. The drop of pressure is less throughout each L.P. turbine to develop the same power as in the H.P. turbine.

2. The amount of steam used in each L.P. turbine is roughly only half that used in the H.P. turbine to develop the same power, as the exhaust from the H.P. turbines divides into two portions, one for each L.P. turbine.

Work done by Difference of Vacuum in L.P. Turbines.—When running at reduced speed with all turbines working (as will be noticed in the example given on page 106), the H.P. turbine carried an initial pressure of 80 lbs. by gauge, and the L.P. turbines each an initial pressure corresponding to 15 vacuum, which worked out, gives an absolute pressure of $7\frac{1}{2}$ lbs., as 15 in. $\div 2 = 7.5$

lbs. vacuum, and 15 lbs. (atmospheric pressure), less 7.5 lbs., is equal to 7.5 lbs. absolute initial pressure. From this it will be apparent that each L.P. turbine, working between an initial vacuum of 15 in. and exhaust vacuum 28 in., develops the *same power* as the H.P. turbine, with an initial pressure of 80 gauge and an exhaust pressure of 7.5 lbs. absolute.

So that,

H.P. turbine	initial pressure	=	80 + 15 = 95 lbs.	absolute pressure.		
„ „	exhaust „	=	(say) $8\frac{1}{2}$ lbs.	„	„	
L.P. turbines	initial „	=	7.5 lbs.	„	„	
„ „	exhaust „	=	(say) 2 lbs.	„	„	

NOTE.—The actual back pressure against the H.P. turbines will be about one pound or so in excess of the L.P. initial pressure, and the back pressure against the L.P. turbines about a pound in excess of the condenser pressures.

Referring to the above figures, it will be evident that each L.P. turbine, working with a total pressure drop of $7.5 - 2 = 5.5$ lbs. only, develops or gives out fully the same total heat drop, and therefore the same developed power as the H.P. turbine, working at a total pressure drop of $95 - 8.5 = 86.5$ lbs. This fact brings out, in a very striking manner, the high heat value of very low-pressure steam when applied to steam turbine practice, and, in the writer's opinion, affords perhaps the greatest contrast of all to reciprocating engine practice.

Pressure Drop in H.P. and L.P. Turbines. —The pressure drop per row is less in the L.P. turbines than in the H.P., as will be seen by referring to the approximate example given on page 28, and the great difference in volume of one pound of steam at high pressure and one pound of steam at low pressure is shown by the following figures :—

H.P. Turbine.

At 158 lbs. pressure gross the volume is 2.81 cub. ft.
„ 160 lbs. „ „ „ 2.78 cub. ft.

The difference = .03 cub. ft. $\left\{\text{for a pressure drop of 2 lbs.}\right.$

L.P. Turbines

At 10 lbs. pressure gross the volume is 37.84 cub. ft:
,, 10.5 lbs. ,, ,, ,, 36.14 cub. ft.

The difference = 1.70 cub. ft. { for a pressure drop of .5 lb.

The above demonstrates the necessity for much longer blades and wider spacings at the exhaust ends of the L.P. turbines.

Number of Rows of Blades.—The number of rows of blades, according to the formula given by Mr E. M. Speakman on page 38, works out as follows:—

EXAMPLE.—Calculate the required number of rows of blades of a turbine if the mean blade velocity is to be 100 feet per second.

RULE.—$1500000 = V_t^2 \times$ number of rows.

Therefore, $\frac{1500000}{V_t^2}$ = number of rows, and $\frac{1500000}{100^2} =$ 150 rows of blades.

EXAMPLE.—The total number of rows of blades in the turbines of a channel steamer is 192. Calculate the mean velocity (V_t) of the blades.

Then, $1500000 = V_t^2 \times$ number of rows.

Therefore, $1500000 = V_t^2 \times 192$,

and $\frac{1500000}{192} = V_t^2 = \frac{1500000}{192} = 781$;

so that, $\sqrt{781} = 88$ feet per second $= V_t$.

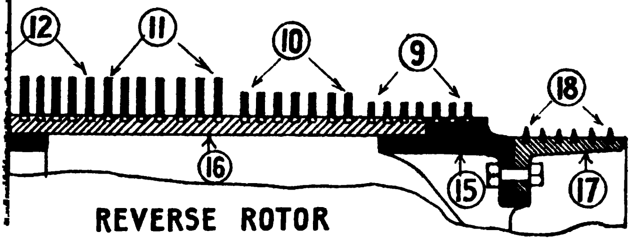
Drop of Pressure.—Between each pair of rows of blades the steam drops a certain amount in pressure, and the approximate drop can be shown as follows:—Suppose the initial pressure in the H.P. turbine to be 140 lbs. gauge, and the terminal pressure at the last row of blades to be, say, 22 lbs., then,

$140 - 22 = 118$ lbs. total drop of pressure in H.P. turbine.

If, then, the H.P. turbine is made up of, say, 60 rows of blades in all, and we divide the total drop by the total number of rows, we obtain the average drop at each row, thus—

118 lbs. $\div$ 60 rows $= 1.96$ lb. average drop per row.

NOTE.—It should be noted that there will also be 60 rows of fixed or casing blades.



s.

expansions. The reverse rotor contains four expansions,

not the same number of blades per row.)

eed 20 knots.

Expansion. (7) Seventh Expansion. (8) Eighth Expansion.
ourth Expansion.
piston. (18) Reverse dummy "fins."
ourth Expansion.

[To face page 28.

Again, take the L.P. turbine, also formed of 60 rows of blades, the initial pressure being, say, 20 lbs. gauge, and the condenser back pressure $1\frac{1}{2}$ lb. (27 in. vacuum), or, say, $2\frac{1}{2}$ lbs. gross actual pressure at last row of L.P. blades.

Then, $20 + 15 = 35$ lbs. gross initial pressure,
and $35 - 2.5 = 32.5$ total drop in pressure in L.P. turbine;
then, $32.5 \div 60 = .54$ of a lb. average drop per row.

It should be noted that this drop takes place simultaneously in each L.P. turbine, as the H.P. exhaust divides into two branches, one for each L.P. turbine. The pressure drop gradually *decreases* from the initial end to the exhaust end of each turbine.

Increasing Steam Velocity.—The velocity acquired by the steam at any given position of the rotor entirely depends on the drop produced by the work done or loss of energy at each row, and as the steam expands in falling in pressure it follows that the area allowed for steam flow must be increased to give equal velocities throughout the turbine. As, however, a Parsons type turbine is composed of a certain number of rows of blades, all of the *same size*, for each expansion, the velocity of the steam at the after rows will be more than that at the forward rows.

To take a practical example. A certain Parsons marine turbine consists, say, of the following:—

H.P.	turbine = 4	expansions,	each	containing	16 rows	= 64	rows.
L.P.	,, = 8	,,	,,	,,	8 ,,	= 64	,,
L.P.	,, = 8	,,	,,	,,	8 ,,	= 64	,,

Total number of rows of blades, 192

Therefore, if the steam velocity at the first row of the H.P. turbine is, say, 250 feet per second, at the last or sixteenth row of that expansion the velocity will be more, as the volume, due to pressure drop, is increased, with the same blade heights and openings. This holds good for each of the various expansion sets or rows, and the increase of steam velocity can be approximately determined by comparing the volume of steam per pound for each of the two pressures, initial at first row, and terminal at last row.

The difference in blade heights is more marked in the

L.P. turbines, as the volume increases very rapidly with fall of pressure in the case of low-pressure steam. The following figures will perhaps make this clear:—

Steam Pressures and Volumes.

Pressure (Gross).	Cubic feet of Steam per pound weight (Steam volume).
210 lbs.	2.15 cubic feet.
195 "	2.31 "
175 "	2.55 "
155 "	2.87 "
135 "	3.26 "
115 "	3.80 "
95 "	4.54 "
75 "	5.68 "
55 "	7.61 "
35 "	11.64 "
15 "	25.84 "
10 "	37.84 "
8 "	46.68 "
6 "	61.20 "
4 "	89.63 "
2 "	172.08 "

Notice that one pound of steam at 210 lbs. gross pressure occupies a volume of 2.15 cubic feet, and one pound at 2 lbs. gross pressure a volume of 172.08 cubic feet.

Pressure Drop, Number of Rows, and Revolutions.—For a given diameter of rotor and required turbine efficiency—

1. A small pressure drop per row produces a low steam velocity, demanding a corresponding low revolution speed, and requiring a large number of rows to absorb the available heat energy of the steam.

2. A large pressure drop per row produces a high steam velocity, demanding a corresponding high revolution speed, and requiring a smaller number of rows to absorb the available heat energy of the steam.

NOTE.—In No. 1 case the revolution speed may be reduced if the diameter of the rotor be increased (as described on page 21) as this will also give the required high peripheral blade speed necessary to still maintain the ratio of $\frac{V_t}{V_s}$ and give constant turbine efficiency.

From 8 lbs. pressure (gross) down to 2 lbs. pressure (gross) the increase of steam volume per pound is very marked, and complicates the design of the L.P. turbines, owing to the necessity for allowing suitable blade heights and openings to prevent rapid increase of steam velocity at the low pressures referred to. The diagram of Blade Heights and Steam Volumes compared, facing page 16, illustrates this point to some degree.

SECTION II.

PRACTICAL CONSTRUCTION, &c.

Arrangement of Turbines.—Originally the turbines were arranged in three sets of increasing size, and the "Turbinia," the pioneer of turbine steamers, was fitted in this way; but latterly the compound systems have been adopted as being the most satisfactory, with the H.P. turbine and shaft in the centre, and one L.P. turbine and shaft on either side. The H.P. turbine exhausts into both L.P. turbines simultaneously, and afterwards to the condensers. Each L.P. turbine is fitted with one reverse turbine at the after end, making five turbines in all.

Drums and Casings.—The blades are not fixed direct to the shaft, but are fitted into steel drums, hollow from end to end, which are shrunk on to the cast-steel "centres," or wheels, and secured by riveted pins (see page 61).

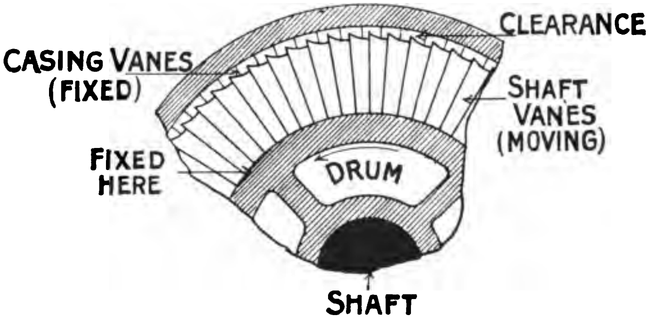
The drums are usually bored out of the solid, a large steel ingot being required for each drum. Recently, however, boiler steel plates, butt strapped and riveted, are being tried, to reduce the cost of production.

These drums increase the leverage of the power exerted in rotating the shaft. The casings, bolted together in two halves, are also fitted with rings of blades; the casing blades project inwards until they just clear the shaft drum by a $\frac{1}{50}$ in. or thereabout, and the shaft blades project outwards until they just clear the inside of the casing with a similar small working clearance.

NOTE.—The drum in which the rotating blades are fixed is called the "rotor" of the turbine.

It should be noted that the rotor drum is of the same

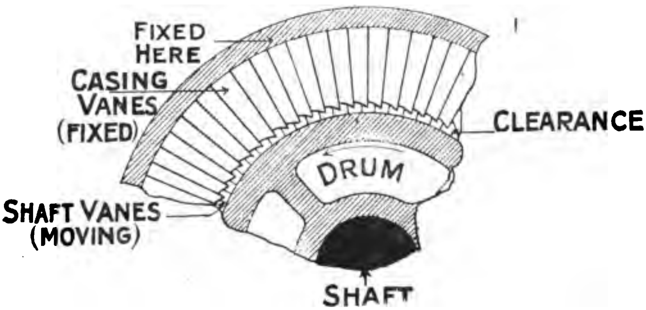
diameter from end to end, the blades only being "stepped" in increasing heights from forward to aft.



Shaft Blades (moving).

View of Rotor Blades.

The economical running of turbine engines is greatly dependent upon the reduction of the clearance spaces to the utmost limit compatible with working conditions; this,



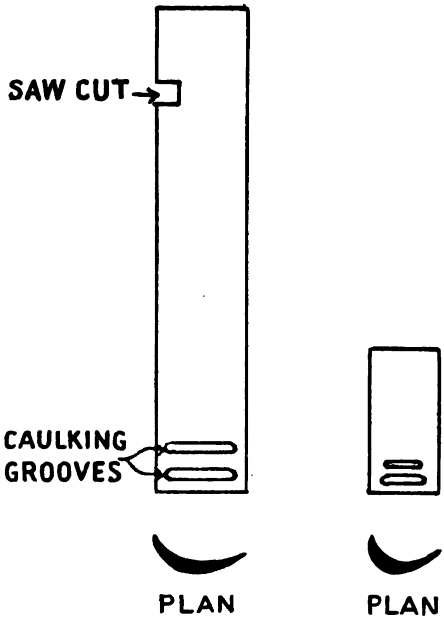
Casing Blades (fixed).

View of Casing Blades.

of course, applies more or less to all steam engines, but in the case of turbines the steam blows continuously through from end to end over the tips of the blades, and this necessarily results in considerable loss of steam which, needless to say, is unavoidable.

Blades.—The vanes or blades, formed of brass, vary in length according to the size and power of the engine, and are fixed by being fitted into a groove turned out of the drum, and held in place by a brass fitting strip caulked well in between each pair of blades. The vanes are curved

in section, and have the edge which exhausts the steam sharper than the inlet edge.



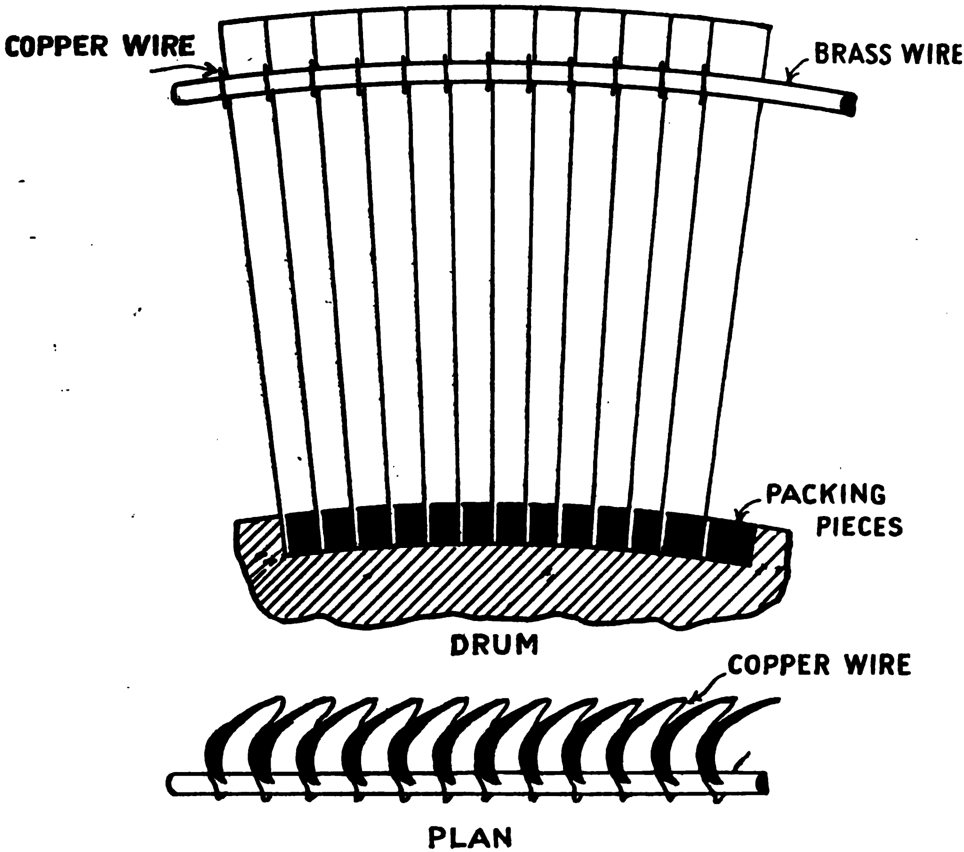
Blades (full size).

The blades do not lie parallel to the shaft axis or at right angles to it, but are placed at a slight angle, the shaft blades being at the opposite angle to those of the casing (see sketch). Each turbine *begins* with a ring of casing blades, and *ends* with a ring of rotor blades.

NOTE.—The various extracts which follow are from a paper read at the Forty-ninth Meeting of the Institution of Engineers and Shipbuilders in Scotland, on 24th October 1905, and entitled "The Determination of the Principal Dimensions of the Steam Turbine, with Special Reference to Marine Work," by Mr E. M. Speakman, Associate Member, &c., and are reprinted here by kind permission of the Council of that Institution. Describing the action of steam in passing through the successive rings of blades, &c., Mr E. M. Speakman says:—

"Restricting attention to the design of the Parsons type of turbine, a few notes on the action of the steam among the blades

may be of interest. Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy; secondly, from



Elevation and Plan of Rotor Blades in Position, showing how secured (full size).

kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

"Fig. 5 gives a sectional elevation of a marine-turbine blading

arrangement, and though this is only for an H.P. cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic, is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is subdivided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The total torque on the shaft is due to the impulse of steam entering the moving blades and to reaction as it leaves them, this process being repeated throughout the turbine.

"Leakage past the revolving portion of the spindle at D is almost entirely prevented by the ingenious form of frictionless packing, shown on a larger scale on Fig. 6. The fine clearances and the sudden increase of section have the effect of alternately wire-drawing and expanding the steam, so that at successive grooves it becomes increasingly difficult for the steam to leak past the fine clearances. In the astern turbines, a radial form of packing, depending on fine tip clearances, must be adopted owing to the difference in expansion between spindle and cylinder. Numerous varieties of these forms of packing exist, some of them being extremely efficient in their action.

"The laws governing the best theoretical velocity of steam and blades are similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines, when it is equal to it.

"Parsons' turbines, however, have been built with V_t, Fig. 7, varying from .25 to .85 of V_s, where V_t represents blade velocity at mean diameter, and V_s the steam speed due to expansion across the row in question. A very usual ratio in electrical work for large units has been $\frac{V_t}{V_s}=0.6$, but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to

proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of caulking piece) of having the same area of openings in so many rows while the steam volume increases so rapidly that it adds to the difficulty of close calculation.

"The potential energy of the steam, corresponding to the 'head' in water turbines, can easily be calculated for given pressure differences.

$$\text{B.Th.U.} \times 778 = \text{Energy in foot-pounds per pound of steam} = e.$$

$$V_s = 8\sqrt{e} = 223\sqrt{\text{B.Th.U.}}\,*$$

"For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of V_t to V_s the greater will be the required number of rows, that is, to obtain the required V_s at each stage a smaller pressure drop per row is necessary, or *vice versa.*

"The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension capable of wide variation without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

$$\text{Mean diameter in inches} = \frac{\text{Blade velocity in feet per sec.} \times 228.\dagger}{\text{R. P. M.}}$$

"To arrive at the corresponding number of rows, the revolutions being given, the ratio of V_t to V_s must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by working out the B.Th.U.'s necessary to give a certain steam speed at each row, see Fig. 8, the available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience, and are similar in use and value to the Admiralty

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* Constant $223 = \sqrt{64 \times 778}$.

NOTE.—$64 = \text{gravity} \times 2$; 778 foot-pounds = 1 B.T.U.

† Constant $228 = (60 \times 12) \div 3.1416$.

NOTE.—60 seconds = 1 minute; $12'' = 1$ foot.

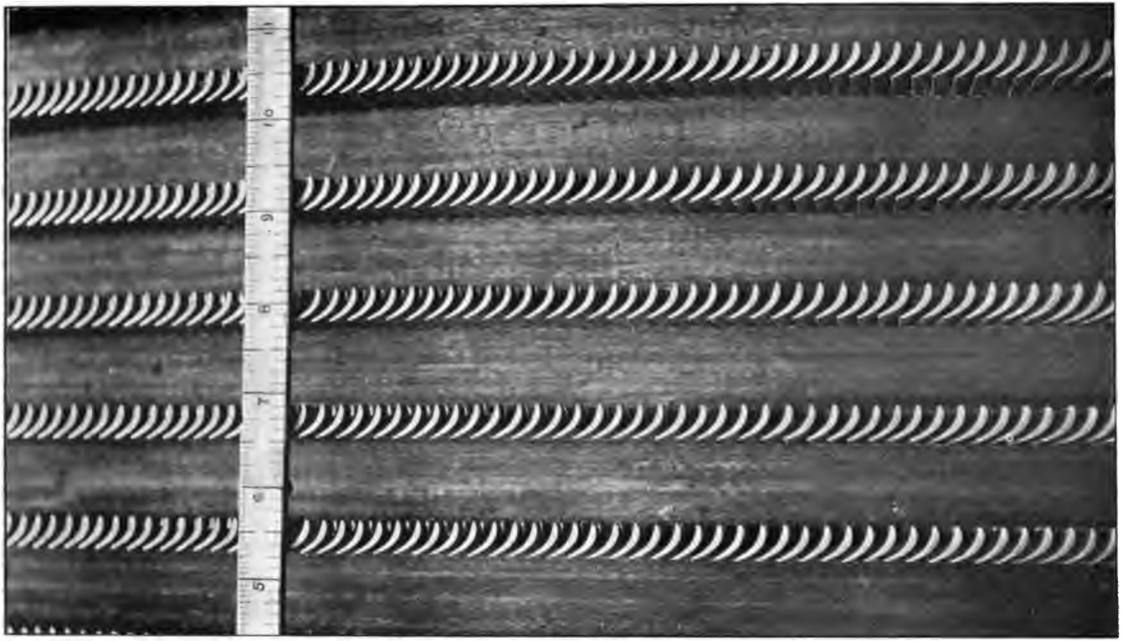
coefficient, that is, while they represent a crude method of doing something that should be done more scientifically, they are very simple and capable of rapid handling. Being, however, based on long and costly experiments, much reticence is observed regarding their publication. Varying, of course, with the steam pressure and vacuum, the number of rows on one diameter would involve an excessive length of turbine and also inconvenient blade heights. It is, therefore, usual to divide the rotor into three or more stages, which have the advantage of shortening the turbine and reducing the number of rows. If $n =$ the fraction of power developed in the first cylinder or barrel, $\frac{N}{n} =$ number of rows in the first barrel, and with the alteration of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels are so altered as to keep, for equal powers and efficiencies—

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant.}^*$$

"The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Co. at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons' blading is, as far as the author is aware, about 375 feet per second in the low pressure blades, and 170 in the H.P. blades of electrical turbines; the lowest speeds used are in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified that this may be at least 3 per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less it may be made. . . .

"In Table II., the vane speeds adopted in various classes of work are given, and the reduction in peripheral speed on account of the propeller reducing the revolutions, and the necessary proportion of blade height modifying the diameter may be clearly

* NOTE.—The "constant" referred to above is given as 1500000.



VIEW OF BLADES LOOKING DOWN ON ROTOR.

NOTE.—The scale of inches shown gives a fair idea of the blade widths and the distance apart of the various rows.

seen. To this combined action is due the fact that only in the faster classes of vessels, or in those small types in which some propulsive efficiency can be sacrificed, is the turbine applicable. In slow cargo steamers, though the revolutions may be high enough, the power required is not sufficient to enable a reasonable blade height to be adopted, and it is this consideration—viz., proportion of leakage over blade tips—that curtails the wider adoption of this type of turbine. For the same low peripheral blade speed, other types of turbine are unsuitable on account of the impossibility of reducing the steam velocity sufficiently without abnormal weight and inefficiency.

"TABLE II.—MARINE WORK.

TYPE OF VESSEL.	Peripheral Vane Speed per Second.		Mean Ratio of V_t—V_s.	Number of Shafts.
	H.P.	L.P.		
High speed mail steamers	70- 80	110-130	·45-·5	4
Intermediate do.	80- 90	110-135	·47-·5	3 or 4
Channel steamers	90-105	120-150	·37-·47	3
Battleships and large cruisers	85-100	115-135	·48-·52	4
Small cruisers	105-120	130-160	·47-·5	3 or 4
Torpedo craft	110-130	160-210	·47-·51	3 or 4

"The smallest size of marine turbine is usually larger than the average electrical turbine as far as power is concerned, and therefore does not meet with the same commercial considerations as the smaller sizes of the latter type. These are not designed for the same internal efficiency as the larger machines, chiefly on account of manufacturing cost, and they do not attain anything like the same efficiency compared with the Rankine cycle.

"Speaking in reply to the discussion on his paper to the Institution of Naval Architects in 1903, Mr Parsons said that, 'for all practical purposes, while the steam is traversing each set

(of blades) as shown, it behaves like an incompressible fluid, just like water would do, as the expansion is very small at each set. The frictional losses and the eddy-making losses would be practically identical within small limits with what they would be with water, and the actual forces would be in proportion to the density of the medium. . . . In the turbine blades themselves, the efficiency is between 70 and 80 per cent.'

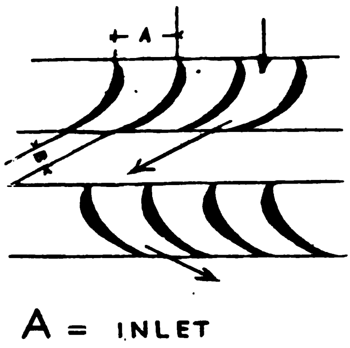


Diagram showing Area through Blades to allow passage of Steam compared to Annular Rotor and Casing Area.

$A \div 3 = B.$

The area B depends on the steam velocity and steam volume, being more for low-pressure steam than for high-pressure steam.

"Using this hydraulic analogy enables us to calculate the number of stages required in a different manner: the 'equivalent head,' due to the steam pressure, may be found, together with that at each row necessary to give the required velocity, from which both the number of stages and the coefficient of expansion at each stage may be worked out.

"In the early marine designs, such as the 'Queen Alexandra' and H.M.S. 'Amethyst,' the turbine drums were all made of the same diameter, and the higher speed necessary on the L.P.'s was got by running at considerably higher revolutions than on the H.P. shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws;

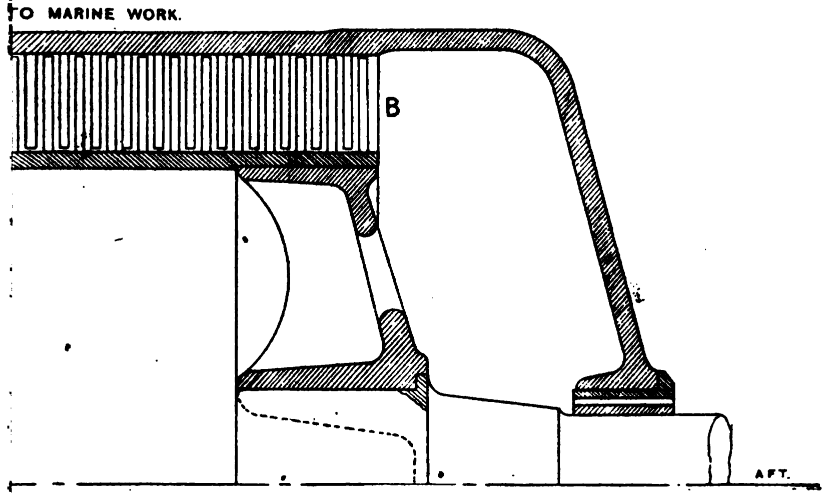
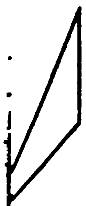
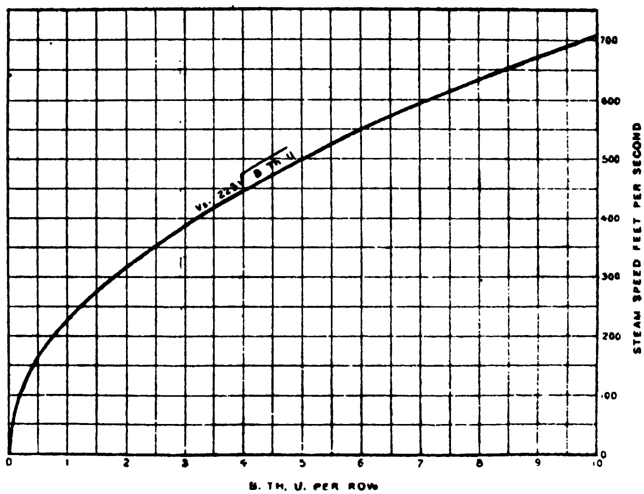


Fig. 8.





[*To face page* 40.

and it is noticeable that in a triple-screw arrangement, the centre screw being right-handed and the wing screws revolving outwards, that the starboard propeller is influenced by the centre one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds and faster at high speeds than the two inner shafts, but exact data as to this, and the possibility of allowing for it in the design, are still wanting.

"In all types of turbines—Parsons', Rateau's, Curtis', &c.—a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the R.P.M. and the diameter than it does with other types. To increase the revolutions, it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

"Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of the steam and the speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume ÷ velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for ½, &c.), then

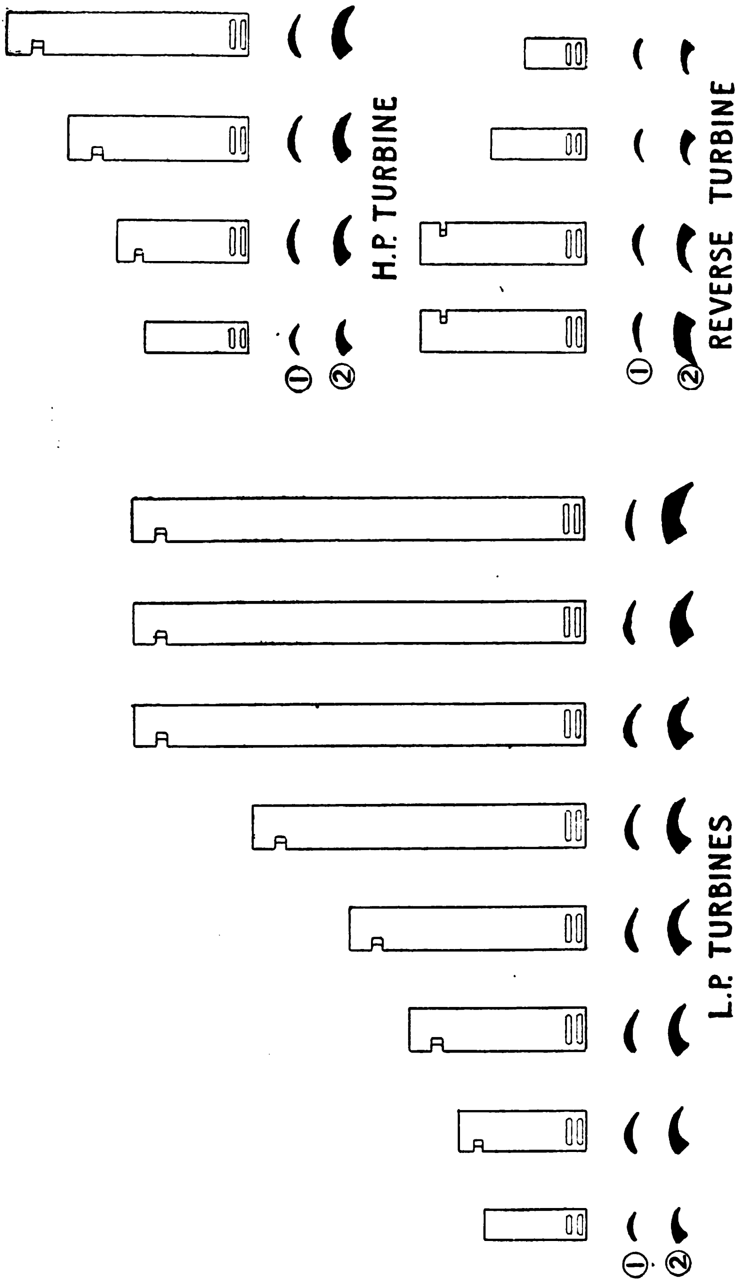
$$\text{Height of blade in inches} = \frac{\text{Clear area in square inches} \times 3}{\text{Mean circumference in inches}}.$$

"The ratio of blade height to mean diameter should not be less than 3 per cent. or more than 15 per cent., because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to

enlarge upon them in this paper. It may, however, be remarked that for $\frac{V_t}{V_s}$ greater than .6 the usual shape of Parsons' section, as shown in Fig. 5, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of strengthening the blade sufficiently, the metal must be placed nearer the exit edge, thus increasing the angle between the face and the back of the exit edge of the blades, and giving, in fact, an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under .6. If, for the present, it is sufficient to use the blade sections and packing pieces similar to those now adopted so generally, in Table III. can be found a list of widths for a given height, and the axial spacing of the rows. While this must be kept down to reduce the length of drum, it must be sufficient to allow for some play in overhauling; and sufficient clearance can be allowed here without affecting the economy. The openings between the blades to allow of the passage of the steam are very important, and must be carefully designed. The actual volume of the steam—not the volume per lb., as found in tables, or the volume due to adiabatic expansion, but the exact volume per lb. at any point along the turbine—must be determined, in order to arrive at the desired adjustment of velocities. It is extremely doubtful whether the present blading arrangements give the best results; greater accuracy of calculation, and consequently improved pressure distribution and efficiency, seem likely to follow the use of a more mechanical blading construction."

Blade Cutting.—The blades are cut to the required length from long strips, and at the same time the machine stamps out the caulking grooves near the bottom end. The saw cut near the top end of the blade is cut afterwards. The saw cut referred to is to allow of the fitting in of the brass binding wire; and in some cases, with extra large blades, double saw cuts for double binding is arranged for.

Cutting of Grooves.—The grooves into which the rotor blades are caulked are turned out of the drum surface, and the sides of the grooves are "toothed" or "serrated," as shown by the sketch, to grip the packing pieces after caulking. Sometimes the rotor grooves are undercut, as shown in the sketch, in which case the serrations are

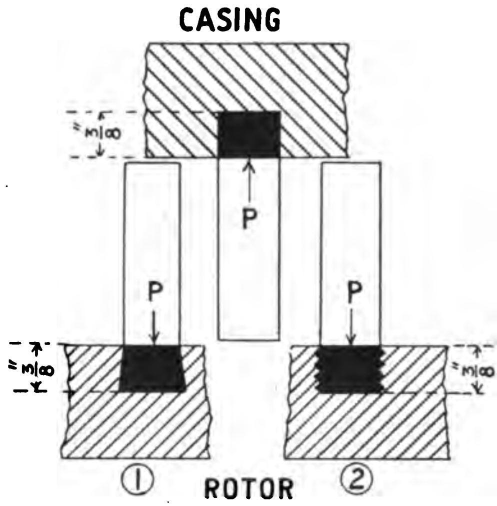


Complete Set of Blades and Packing Pieces (about half size).

1. Plan or edge view of Blades. 2. Plan of Packing Pieces.

NOTE.—The above are from the turbines of a steamer of about 5,000 I.H.P.

omitted, although the parallel grooves with serrations are now usually adopted. The adjustment of the tool requires



Methods of Securing Blades.

(Black Section shows Packing Piece caulked in position.)

1. "Dovetail" Method.
2. "Serrated" Method (parallel groove).

NOTE.—No. 2 is the usual method adopted for the rotors, while for the casings the parallel plain groove as shown above is commonly employed.

accuracy within a few thousandths of an inch, and even the wear of the tool itself has to be allowed for.

The grooves in the casing or rotor cylinder are not usually serrated, as the blades are fixed and do not revolve.

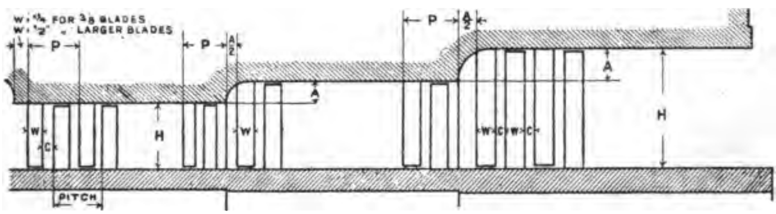
Rotor Blading.—In the work of fitting in the blades, a fixed "stop" piece (which sets the correct blade angle) is first inserted in the rotor grooves and held in position by means of a steel wedge, the first blade and packing piece are then inserted, and after a number are placed in position loosely by hand, the set are tapped up with a hammer and tool; after one ring is filled up in this manner the caulking in of the packing pieces is then proceeded with by means of a specially shaped tool and light hammer, two blows of which are usually sufficient to secure each packing piece. Each expansion requires a separate caulking tool. The

next stage is that of binding, which is done by a brass wire, which fits into a "saw drift" near the outer end of each blade, and when fitted forms a brass ring extending round the circle of blades near their outer end. In the extra long L.P. blades of the Cunard steamer "Lusitania's" turbines, triple binding wires are being employed. After this, the "lacing" is proceeded with, which is effected by means of fine copper wire lashed round each blade, and round the

TABLE III.—STANDARD BLADING DIMENSIONS.

Height (H)	1″	2″	3″	4″	6″	8″	10″	12″	15″	18″	21″	24″	30″
Width (W)	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{5}{8}$″	$\frac{5}{8}$″	$\frac{3}{4}$″	$\frac{3}{4}$″	1″	1″	$1\frac{1}{8}$″	$1\frac{1}{4}$″
Pitch (P)	$1\frac{1}{8}$″	$1\frac{1}{8}$″	$1\frac{1}{4}$″	$1\frac{5}{8}$″	$1\frac{3}{4}$″	$2\frac{1}{8}$″	$2\frac{1}{4}$″	$2\frac{1}{2}$″	$2\frac{5}{8}$″	$3\frac{1}{8}$″	$3\frac{1}{4}$″	$3\frac{5}{8}$″	4″
Axial clearance (C) ...	$\frac{3}{16}$″	$\frac{3}{16}$″	$\frac{1}{4}$″	$\frac{5}{16}$″	$\frac{3}{8}$″	$\frac{7}{16}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{9}{16}$″	$\frac{9}{16}$″	$\frac{5}{8}$″	$\frac{11}{16}$″	$\frac{3}{4}$″

Note.—While the above represents general practice, it is obvious that such a table is largely arbitrary.

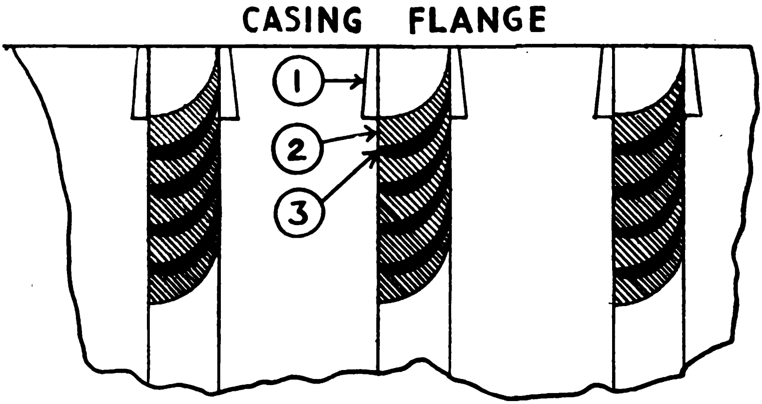


Blade Dimensions.

brass binding of each wire (see sketch). The copper wire is connected up in short lengths with the ends of each twisted together. The blades are then turned up by machine at the tips for correct clearance adjustment, on which the economy and efficiency of the turbine greatly depends. Finally, the whole—blades, binding wire, and copper wire—are brazed by means of *silver* solder and a blow flame. The blade surfaces are afterwards "cleaned up" by means of a compressed air blast to remove the "burr" left by the turning up of the tips by machine.

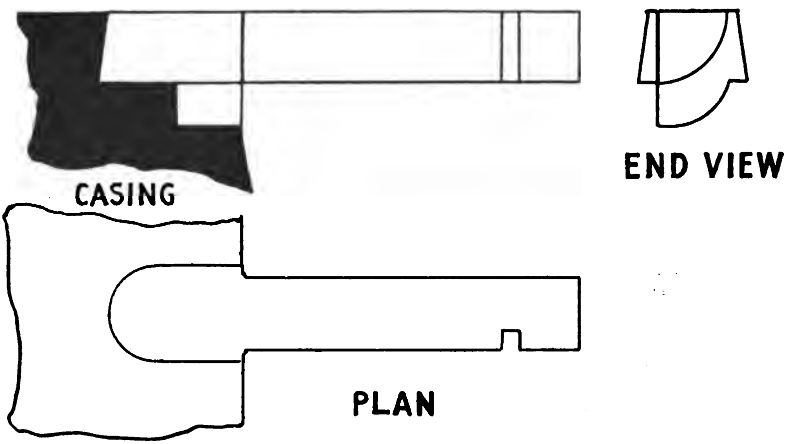
Casing Blading.—Brass stoppers are caulked into recesses drilled out of the flange face of the upper and lower half casings at each side of the diameter. These stoppers set the angle of the first packing piece and blade fitted into the grooves of the casing, two stoppers being

employed to each half casing, therefore four in all to one complete row or ring of casing blades. Each pair of stoppers bear together face to face when the two half



View of Lower Half Casing (looking inside).

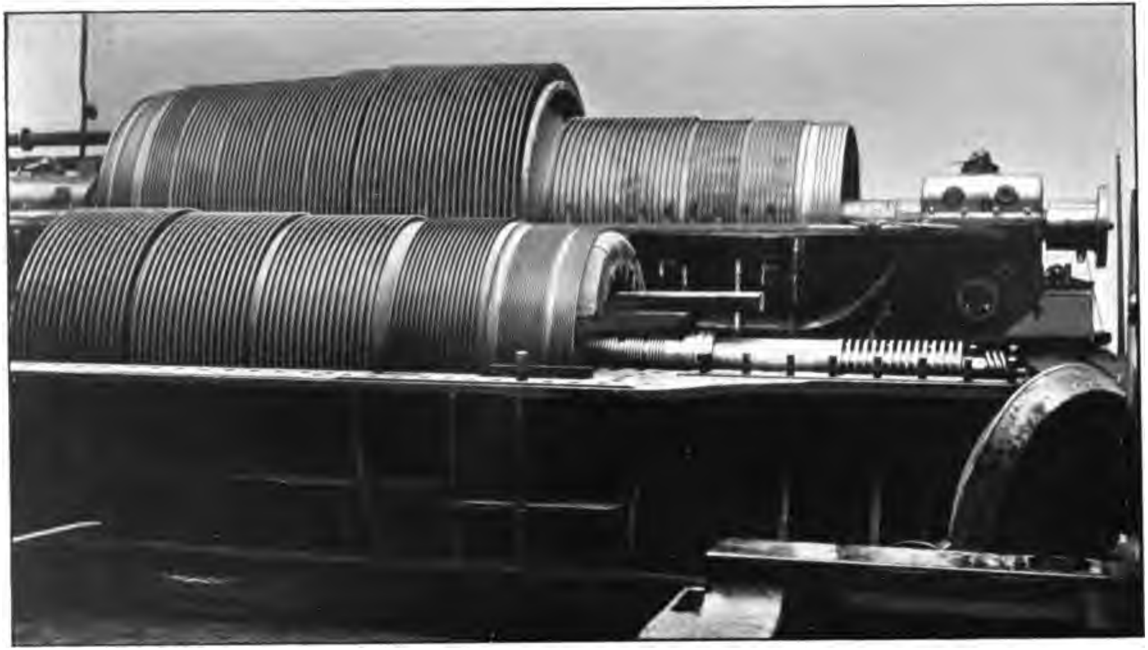
1. Blade Stopper. 2. Packing Piece. 3. Blade.



Casing Blade Stopper.

NOTE.—The stopper is fitted into the casing flanges top and bottom, and sets the angle of the blades.

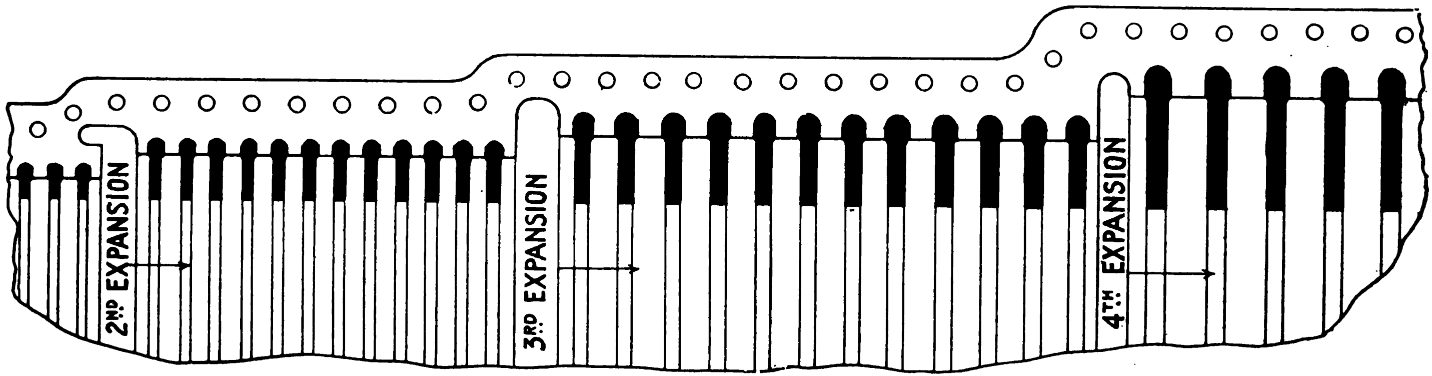
casings are bolted down. The length of the stoppers exactly corresponds to the length of the blades for which they are intended, and it should be noted that the stoppers are left in position permanently. The blading of the



VIEWS OF H.P. AND L.P. TURBINES.

Showing Ahead H.P. Dummy, and the four H.P. Expansions, also the L.P. Ahead and Reverse Dummies, and the eight L.P. Ahead Expansions, and four Reverse Expansions.

NOTE.—Each H.P. Expansion contains sixteen rows of blades, and each L.P. Expansion contains eight rows of blades.



Part of Turbine Casing showing Blade Stoppers in position.

casing is carried out in the same manner as that described for the rotors, the various stages being in every way similar.

NOTE.—No permanent stoppers are fitted in the rotor grooves.

For good turbine efficiency results it is imperative that the angles, pitch, and position of the blades be as nearly mathematically correct as possible, any tendency to unequal circumferential pitch or angle (the blading being then known as "staggered") resulting in loss of efficiency.

The operation of "blading" being carried out entirely by hand labour, forms the most costly item in turbine construction.

Stresses on Blades, &c.—It should be observed that the casing blades are subjected chiefly to a bending stress, due to the action of the steam in striking the blade surfaces, whereas the rotor blades, in addition to a bending stress, are also subjected to a severe tensile stress set up by the centrifugal force of the rotor when revolving, and tending to throw out bodily the blades and packing pieces. To resist this, the grooves cut in the rotor drum are either parallel and serrated, as shown in the sketch on page 44, or are undercut, as shown in the sketch facing page 62, the casing grooves being merely parallel and without serrations, this arrangement having been found to give perfectly satisfactory results.

The little stamped-out grooves shown near the bottom of the blades (both casing and rotor) allow of the packing piece being caulked in to lock the blades firmly in position.

Each "stopper" is of the same length as the corresponding blades. Regarding the materials employed for the blades, and clearances allowed, Mr E. M. Speakman says :—

"The material of which blades are usually made is a mixture of cheap brass containing about 16 per cent. of copper and 3 per cent. of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about 98 per cent. of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 per cent. of

copper and 20 per cent. of nickel has been adopted, and this is undoubtedly the best blading material existing. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal used, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

"It seems probable that the usual caulking piece now adopted will be discarded in favour of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing, and soldering process at the tip, a similarly machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained; at the same time considerably reducing the cost of manufacture, and the risk of blade stripping.

"The chief causes of the latter may be set down to bad workmanship in fixing the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design, or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to generally occur from preventible causes. Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length that fatigues the blade material and causes the loss of blades without any fouling at all.

"Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of the thrust block necessary. A margin must be allowed here, and the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring diameter D and the spindle, Fig. 5, C, plus the end pressure on the blades. For the diameter D to give the required annulus, as well as that of the propeller, the effective thrust must be carefully calculated; and experience shows that there is a drop in steam pressure varying from 10 to 15 lbs. per square inch between the pipe inlet to the H.P. receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very largely, and may be modified according to the pressure and the clearance allowed—say a 7-1000th

to a 15-1000th of an inch in electrical work, and rather more in marine work.

"The dimensions of the astern turbine are arrived at in the same manner as those of the ahead, the efficiency being largely sacrificed on account of weight and space; generally, the mean diameter is made practically the same as that of the H.P. drum.

"To a large extent, the inferior manœuvring capabilities of the earlier turbine steamers were due to insufficient astern power.

"It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working. In electrical turbines where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. The stresses due to centrifugal force are very low in the Parsons turbine, and except in occasional L.P. barrels do not exceed about 7,500 lbs. per square inch, while at the H.P. end they are usually under 2,000.

"The pressure on the bearings in a turbine is only due to the weight of the spindle, plus the negligible addition in marine work of that due to any gyroscopic action; it may be taken as from 80 to 90 lbs. per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure × velocity does not exceed 2,500-2,700. In land work, 50 lbs. × 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperature should not exceed from 140° to 150° F., though the writer has known of 190° F. being used without trouble. In marine turbines this temperature is usually much lower. Rigid bearings are used for marine spindles, not the flexible type adopted in land work.

"Space does not permit of more than passing reference to cylinders; but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders, with heavy flanges on the centre line, distort in a very curious fashion when heated with their axis horizontal, and measurements taken off a hot cylinder on a surface plate with micrometer gauges reveal some very remarkable facts. When working, the temperature along the cylinder falls possibly from 400° to 100° F. in a distance of 6 or 8 feet, and, unlike the reciprocating engine, this remains

constant; the radial expansion is consequently more at one end than the other; while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

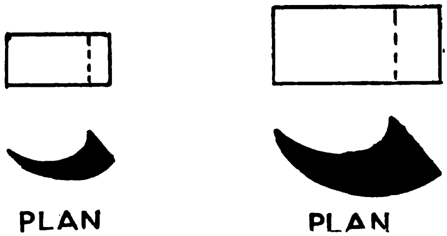
"The longitudinal expansion when hot is often very marked, and in all turbines necessitates provision for the resultant movement at one end. In marine work the after end of the cylinder is secured to the vessel, the engine seating also performing the function of a thrust-block seat, while the forward end slides forward, taking with it the entire shafting. The thrust block is at the forward end of the cylinder, and also performs the duties of an adjustment block for setting the longitudinal clearances, to do which generally necessitates uncoupling the shafting abaft the turbine.

"The difference in expansion between the cylinder and spindle, from the thrust block to the dummy ring, may be the cause of serious difficulties in large marine turbines, unless the closest attention is paid to this feature in the design; and 'warming up' with these large cylinders needs possibly even more care than is essential with large piston engines.

"On shipboard, the turbine cylinders are practically under one's feet, and the radiation from them is very unpleasant, especially if there is any leakage from the glands. To all who are responsible for the lagging of cylinders and the system of ventilation in turbine engine-rooms I would call attention to the possibility of their having to stand a watch of four or six hours on the top of the H.P. cylinder, such as is the case in the 'Eden' or 'Amethyst,' the heat in the latter vessel being almost unbearable. With reciprocating engines, one stands on a comparatively cool lower platform with the cylinders overhead, and with some chance of the hot gases rising clear, but in naval turbine work under a low deck this point has not met with adequate attention.

"In the course of operation, more especially in marine work where no superheaters are used, there is a distinct tendency for the turbine to be supplied with wet steam, the effect of which on the economy is very marked. Experiments that have been made show that the percentage increase in consumption is about twice that of the moisture in the steam. For instance, with 2 per cent. of moisture in the steam at the first row, the consumption is increased about 4 per cent."

Packing Pieces.—The small packing pieces which are caulked in between every pair of vanes are made of brass, and vary in size according to the position occupied by them



"**Packing Pieces**" **(full size).**

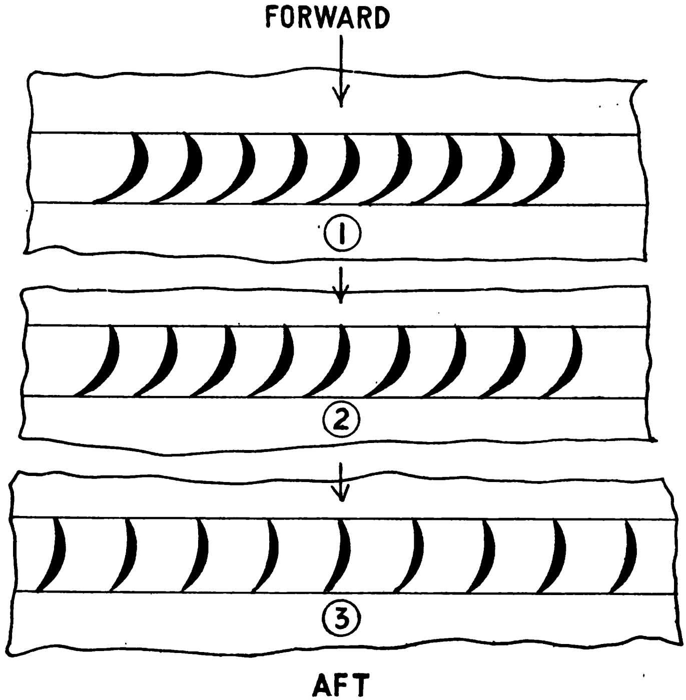
in the turbine, being small at the forward or higher pressure end, and larger at the after or lower pressure end (see sketch).

Clearance.—The clearance allowed between the dummy rings and grooves is very small, and is usually set for between $\frac{25}{1000}$ and $\frac{50}{1000}$ of an inch. Should, therefore, any appreciable change of position longitudinally of the rotor take place, the rings and grooves would come into contact, and damage result. Again, should the bearings wear down or run out by heating up, the rotor blades would foul with the casing, and the casing blades foul with the drum, resulting in stripped blades. The circumferential clearance at the blade tips varies from about $\frac{1}{50}$ in. ($\frac{20}{1000}$) at the H.P. end to about $\frac{1}{25}$ in. ($\frac{40}{1000}$) at the L.P. end.

Expansion Clearance.—At each change of expansion a clearance space is arranged between the last ring of blades of one set, and the first ring of the next set (see sketch) to allow of drop of pressure.

Blade Variation.—In the largest size of blades which are fitted on the after or low pressure end of each turbine, the curve and pitch is varied so as to really constitute three expansions, although the *height* of the blades is the same. The sketch (page 53) shows the variation of blade curvature and pitch, which is arranged to allow for the increasing volume of steam passing through the last section. If the last set of blades is made up of, say, 24 rings, all of the same height, then 8 rings are as of the 1st set, 8 of the

2nd set, and the remaining 8 as of the 3rd set. Observe that the blades are less curved aft, also that the pitch is



Variation in Blade Angles and Pitch in last three L.P. Expansions.

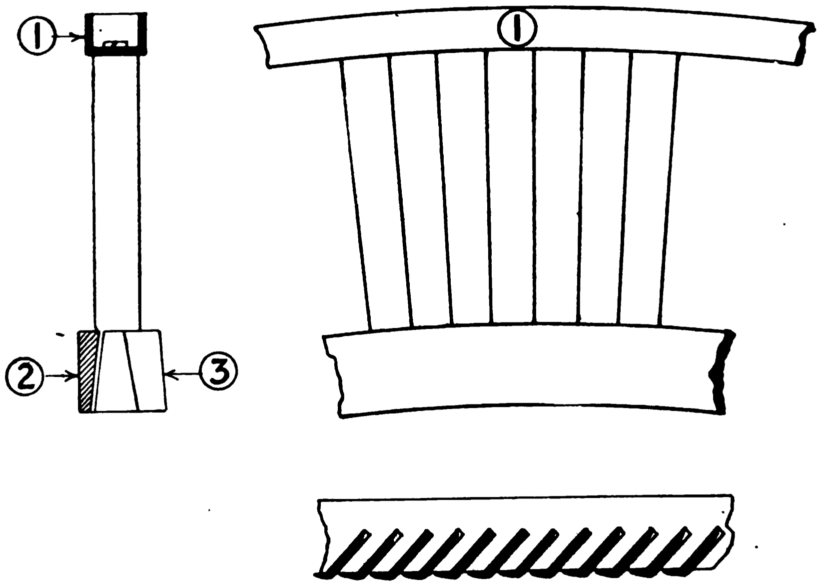
1. Sixth Expansion. 2. Seventh Expansion. 3. Eighth Expansion.

increased, which results in a smaller number of blades per ring. Theoretically each successive ring of blades throughout the whole turbine should be graded in this way.

NOTE.—In some cases the exit openings of the first half of the rows of each of the first few expansions have been reduced, which practically gives two expansions to what was formerly only one: this alteration increases the velocity of the steam passing between the rows with the restricted openings. A specially designed "closing-up" tool is employed to reduce the blade openings as described.

Improved Method of Blading.—Messrs Willans & Robinson have devised and applied successfully a new and improved method of blading which reduces labour and which

also ensures greater accuracy of blade spacing, the latter point being one of considerable importance. The blade roots are fitted into two solid half rings, which are accurately divided off by machine cuts, and thus give uniform adjustment to the blade pitches and angles throughout; at the outer ends or tips the blades fit, by means of a tang, into a channel-shaped brass ring, or "shroud," as it is called. The blading is therefore completed independently in two or



Willans & Robinson Blading.

1. Channel Ring Shroud. 2. Caulking Ring.
3. Tapered Packing Section Ring.

more sections before being fitted on to the rotor or into the casing. The channel-shaped shroud can be adjusted to reduce the tip clearance, as should fouling occur the channel ring would wear away and give its own clearance, or would perhaps bend over, thus protecting the blades and eliminating the danger of blade stripping.

It will be noticed from the foregoing description that in this system of blading no separate "packing pieces" are required, and that the brass wire "binding" and upper wire "lacing" are only required in the case of very long

blades. It seems extremely probable that this method will come into general use, as the advantages are many, and the disadvantages few.

NOTE.—The above method of blading is being adopted in the upper half casings of the "express" Cunarders, and the ordinary system, as previously described, in the rotors and lower half casings.

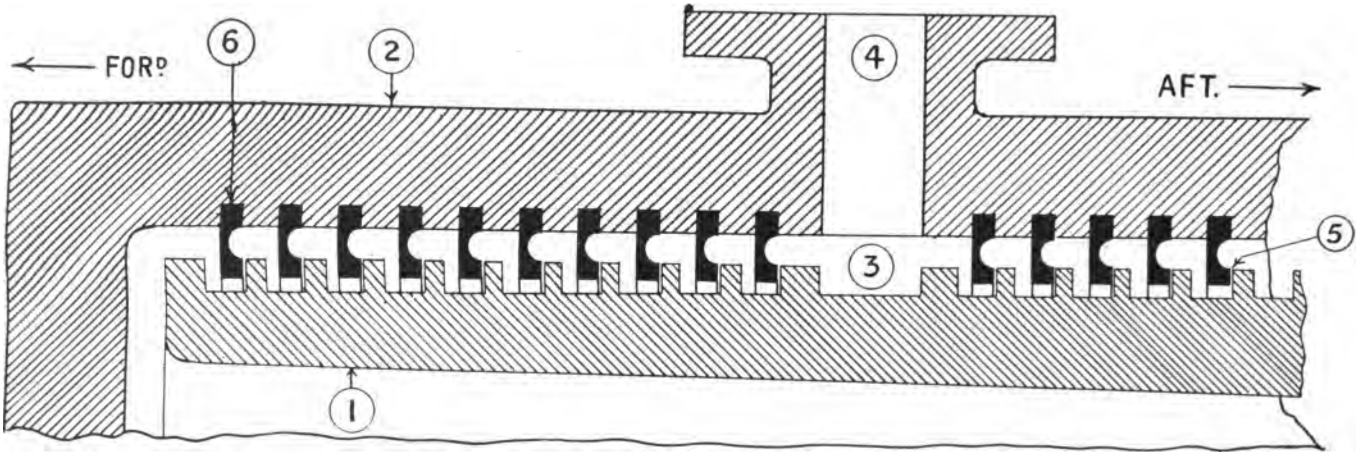
Reverse Turbines.—The reverse turbines consist of a set of smaller blades placed in the after end of the L.P. turbine casings. The angle of these blades is the same as the ahead, but the curve is on the other side, and as the steam is admitted at the *opposite* end of the turbine casing, the direction of shaft rotation is reversed. Both the ahead and reverse turbines exhaust into the same central exhaust passage leading to the condenser. The reverse turbines are also fitted with dummy pistons.

The reverse casing is fitted with small drain holes (in the lower half), one hole for each expansion (at the end row of each). The water so drained off passes into the exhaust space, and then into the condenser.

In the latest types of turbine steamers the size of the reverse turbines has been considerably increased over those fitted to the "King Edward," with the gratifying result that no trouble whatever is experienced in quick stopping and reversing, the vessel being brought to a stand within a very reasonable time indeed—in fact, the stopping and reversing can be effected in less time than that taken with ordinary reciprocating engines.

NOTE.—The reverse rotor is often made of the same diameter as the H.P. rotor, and the reverse power developed equal to about one-third of the ahead power.

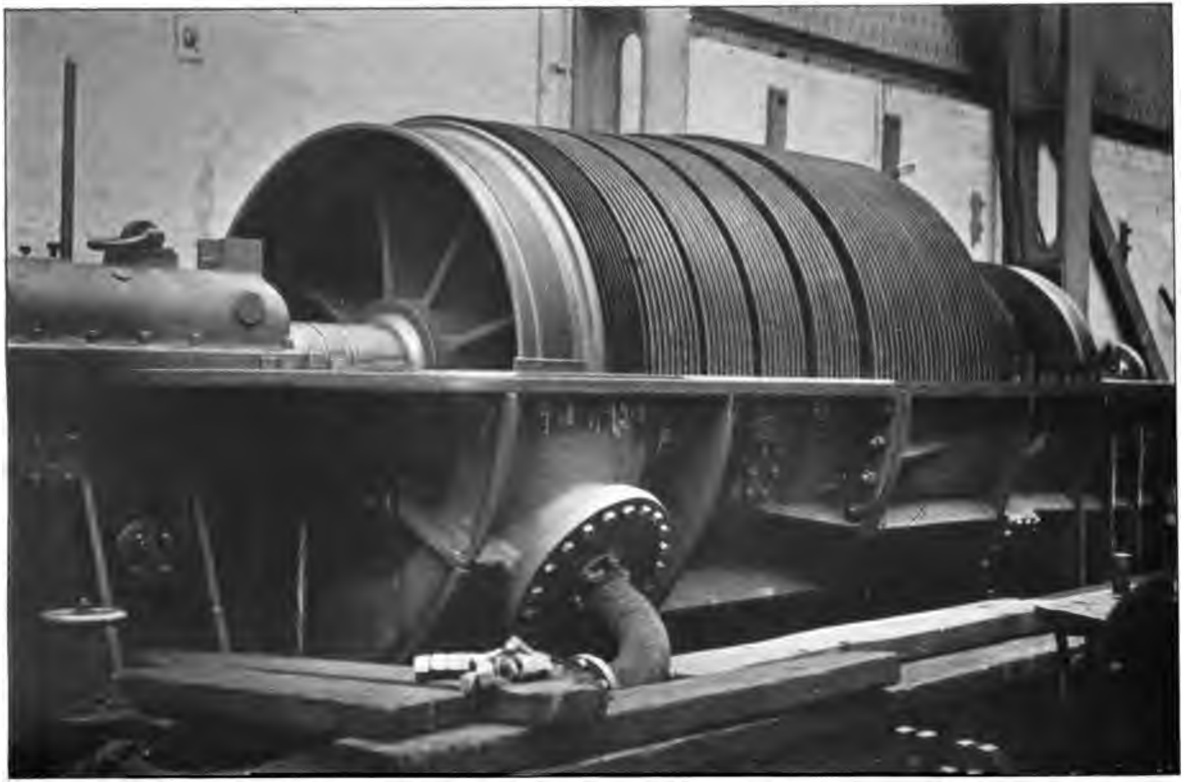
Ahead "Dummy" Pistons.—These pistons, fitted forward and bolted on to the metal of the rotor, are of rather less diameter than the rotor drum (see sketch), and are supplied with a large number of small grooves into which fit a corresponding number of fixed brass rings which are bedded into the dummy casing. These brass rings are hollowed out (or "undercut") on the after side to reduce the



Sectional View of Ahead Dummy.

1. Dummy Piston. 2. Dummy Case. 3. Leak-off Space to Third or Fourth Expansion of same Turbine. 4. Leak-off Port. 5. Clearance of about $\frac{30}{1000}$ of an inch (.03 inch) when heated up. 6. Undercut Brass Rings bedded into Dummy Case.

NOTE.—These rings (often 24 in number) are formed of $\frac{3}{16}$-inch brass at a pitch of about $\frac{1}{2}$ inch.



VIEW OF L.P. TURBINE AND REVERSE TURBINE IN LOWER HALF CASING.

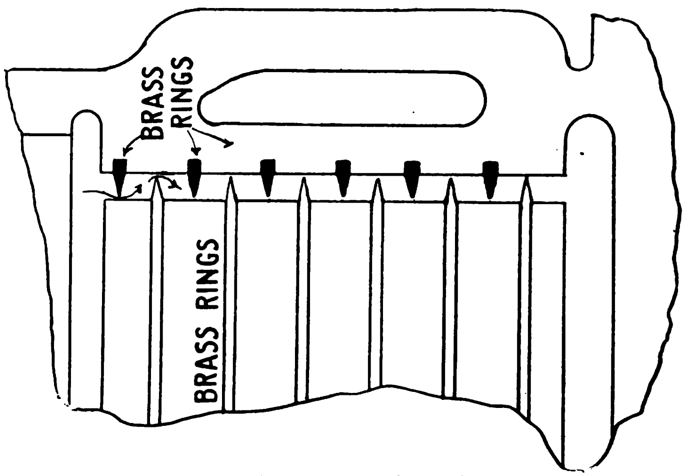
Showing (left) "Wheel" or "Centre" in Lower Half Casing, Ahead Dummy Piston, Ahead Expansions (eight in number), Reverse Expansions (four in number), and Reverse Dummy Piston.

NOTE.—The last three Ahead Expansions have blades of equal height, but of wider spacing, and the last two Reverse Expansions have blades of equal height, but of wider spacing. The steam pipe shown is only that fitted in the shop for "steaming" purposes.

surface in "grinding up" (see sketches), and also allow of the leaking steam being wire-drawn. When the "dummies" move round, the brass rings which fill up the grooves act as baffles to the passage of steam, and in addition to this the dummy piston assists to balance the steam thrust on the blades which acts aft, that is, in the opposite direction.

NOTE.—The clearance allowance for wear forward of the dummies is only about $\frac{50}{1000}$ part of an inch at most, and this is set by the adjusting half ring fitted in the bed of the lower half of the thrust block, referred to elsewhere (page 68).

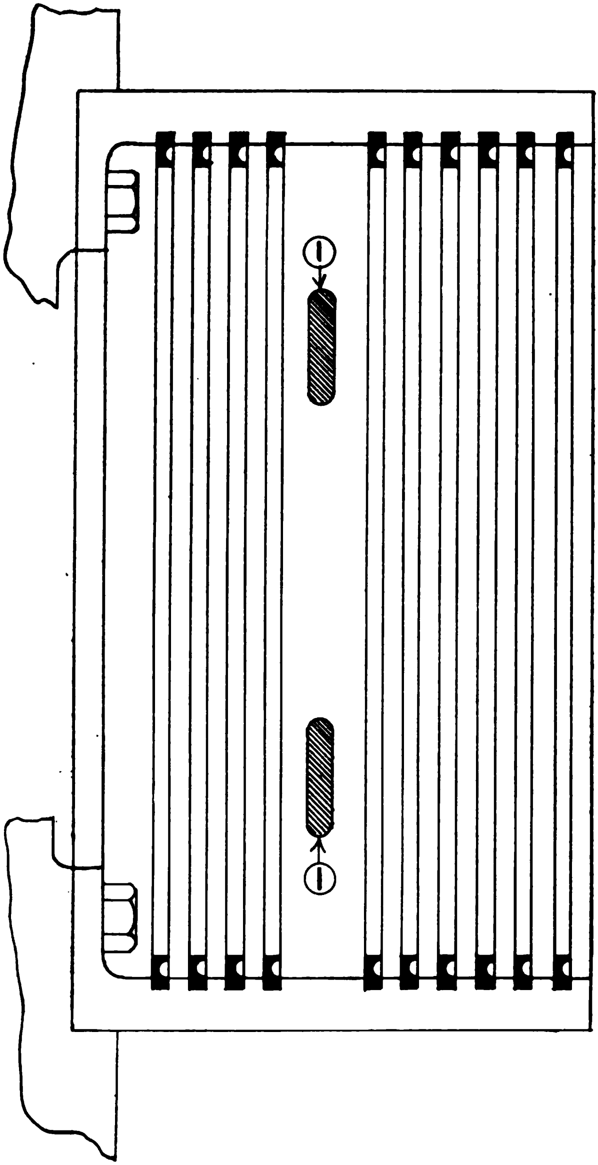
Reverse "Dummy" Pistons. — The dummy pistons of the reverse turbines are slightly different from those of the ahead, as instead of the brass rings of square section bedded into the casing, rings of a tapered or wedge-like section are adopted, the point of the wedge just clearing the surface of the dummy piston. Similar "radial fin" section rings are fitted to the dummy, and just clear the



Reverse Dummy showing "Fins."

NOTE.—The "radial fins" are usually pitched about 1 inch apart, and are formed of $\frac{1}{8}$-inch brass.

casing. This is said to allow better for the expansion at that end of the casing and drum. A number of these rings are fitted to each reverse dummy, sometimes 12 in all.



Ahead Dummy Casing.

I, I.—"Leak-off" Ports.

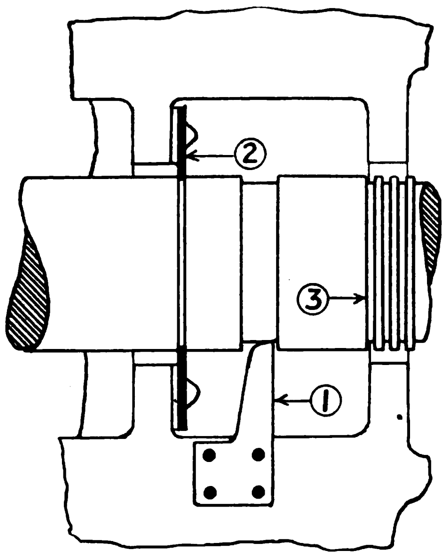
NOTE.—Usually not less than 20 of the small brass rings shown are fitted to each ahead dummy. The rings are formed of $\frac{3}{16}$-inch brass at a pitch of about $\frac{1}{2}$ inch.

Dummy "Leak-off."—The ahead dummies are sometimes supplied with a "leak-off" from the space between the sets of rings to the 3rd expansion of the same turbine.

Any steam which finds its way past the inward set of rings passes away by the "leak-off" ports and pipe to one of the other expansions where the pressure is less (see sketch).

The "dummies" prevent steam leakage at the high pressure ends of the rotor and act so as to produce (by means of wire drawing off the escaping steam) a "water seal" by the resulting condensation.

Dummy Clearance Gauge.—A small flat plate, called a "finger plate," is pinned down to part of the lower casing frame at the forward end, and the edge of the plate



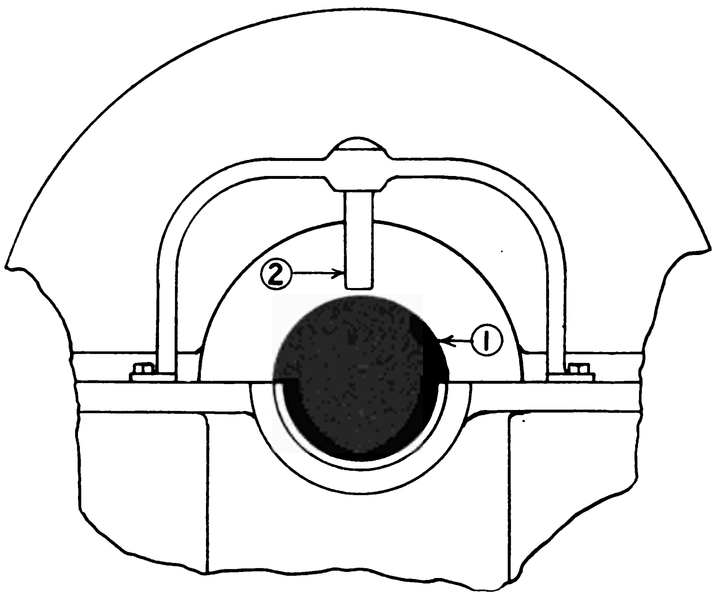
"Finger-plate" Gauge.

1. Finger Plate. 2. Oil Deflector. 3. Steam Gland Rings.

projects against a collar on the turbine shaft, so that by means of a feeler or wedge the clearance between the two can always be known. This clearance is noted when the turbine is completed, and a record kept, so that any change

of clearance due to wear fore or aft can at once be noted by future testing. Longitudinal change of position may occur through wear of the thrust block rings, and to take up this (as mentioned elsewhere) a screw is fitted on the top half of the thrust block cover with an adjusting nut, and half brass rings fitted for adjustment of the lower or ahead portion of the thrust. A set of these rings of varying thickness is supplied as space gear, and can be inserted as required so as to maintain the same dummy clearance.

"Bridge" Gauges.—An appliance called a "bridge gauge" is used at each end of the rotor to test for possible wear down, which, as will readily be understood, is a matter



"Bridge" Gauge.

1. Rotor Spindle. 2. Steel Pin.

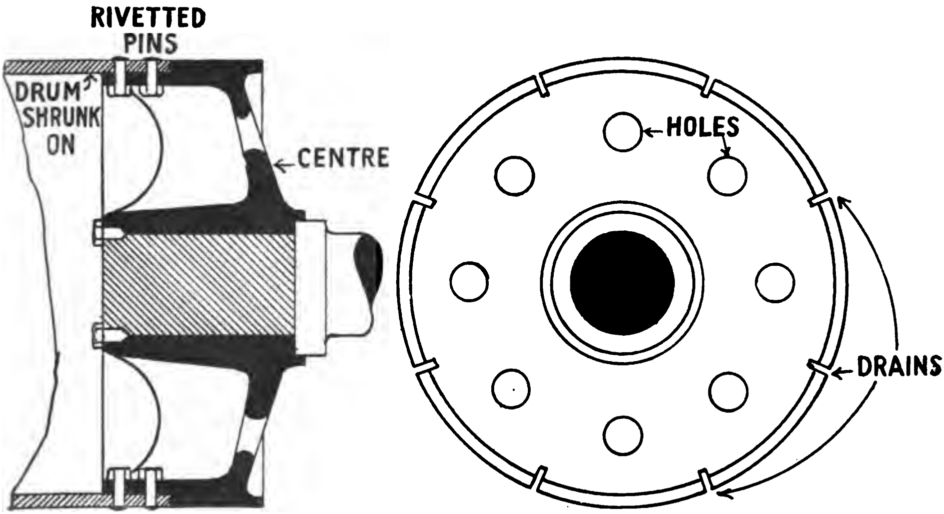
NOTE.—A record is kept of the original clearance between the gauge pin and the shaft at each end, and by reference to this record any subsequent wear down can be at once noted on testing.

of paramount importance. The gauges merely consist of a piece of bent round bar iron with a centre boss through which is screwed and riveted a hardened steel pin with a

flat end. The bridge piece is pinned down to the lower half casing just outside the gland (see sketch), and the pin is adjusted to a known clearance, say $\frac{50}{1000}$ of an inch, from the top of the shaft, so that by applying this test arrangement at intervals any wear down of the rotor can be detected by means of "feelers" placed between the flat end of the pin and the upper surface of the rotor spindle (see sketch).

NOTE.—From the foregoing descriptions of the two appliances, the "Bridge gauge" and "Finger plate," it will be obvious that the wear down and wear forward or aft of the rotor is under constant observation, and the risk of any serious accident, such as dummy ring stripping or rotor blade stripping, almost eliminated. The smallest possible change of rotor position can be accurately measured by the finely graded "feelers" used in testing the clearances shown by the gauges.

Rotor Construction.—The "rotors" are made up in separate sections, which are heated and shrunk on to



Rotor "Centre," or "Wheel."

each other. The sections referred to are spindles, "centres," and drum; and, in the case of the L.P. rotors, the reverse

drum, which is expanded on to the main or ahead drum, at the after end. The "centres" are further secured to the drums by means of pins tapped through the rotor shell and then riveted over, while the spindles are also further secured by pins tapped half into the spindle and half into the "centre" (see sketch facing page 62).

NOTE.—In long rotors an extra wheel is fitted inside at the middle of the length, and in some cases two inner wheels are fitted ("express" Cunarders).

Spindles.—The "centres" or "wheels" are shrunk on to the spindles (which are heated up by a blow flame, either gas or oil, for the purpose), and are further secured by pins tapped half into the spindle and wheel, as shown in the large sketch facing page 62.

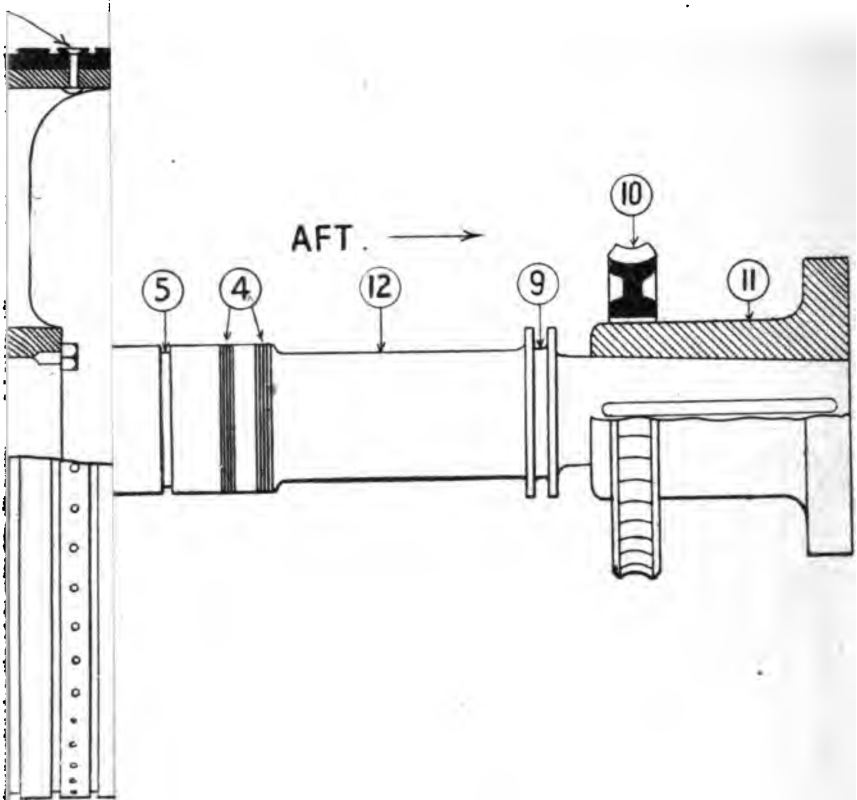
Referring to the sketch, the spindles forward are turned out for steam gland collars, "finger plate" groove, oil-deflecting ring groove, V cut oil grooves to throw off the oil by centrifugal force, main bearing, and thrust collars.

At the after end the spindles are similar, with the omission of the thrust collars and the addition of a double collar groove for a special oil-deflecting ring to keep the oil from escaping out of the bearing casting.

The various oil-deflecting rings, shown at the after end of the forward bearing and the forward end of the after bearing in the sketch facing page 62, are for preventing the admission of any oil into the turbine casing, which is a matter of importance for the safety of the turbine and the life of the boilers, not to mention waste of oil.

"Static" Balancing of Rotors.—The rotors are carefully balanced by means of two truly levelled rails, or "knife edges," on which the rotor spindles are placed. The rotor is then moved back and forward on the knife edges, and the balance adjusted by either pinning on pieces of metal on the lighter parts, or by removing (chipping) some of the metal off the heavy parts.

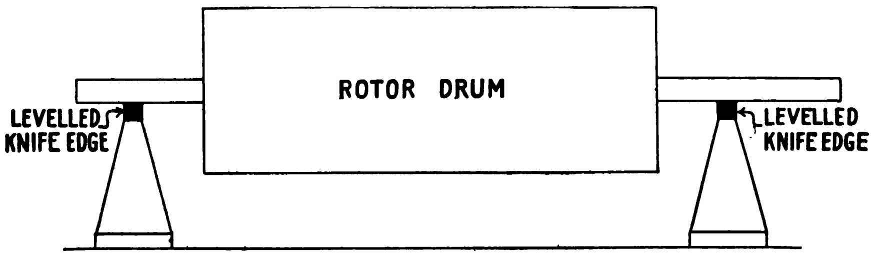
"Dynamic" Balancing of Rotors.—The turbine is balanced dynamically by one of two methods—(1)

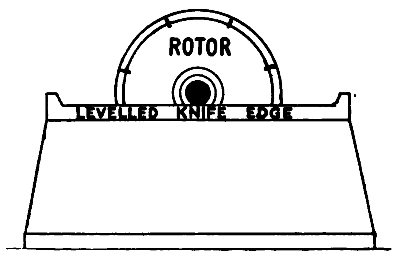


-t Secti
es are un
deflecting Turning wheel.
eted pins.

[To face page 62.

By being revolved by steam acting as under working conditions, and tested by fixed indicating pointers or pencils. (2) By some suitable arrangement of spring bearings in which the rotor is placed and rotated by some external power, such as an electric motor coupled up to the spindle, and as before a pointer indicates the vibrations on a piece of diagram paper by a more or less "waving" line. The latter method is the most accurate, as the rotating force acts from the centre outwards, and, if anything, tends to magnify any lack of balance or difference in weight which the rotor





"Static" Balancing of Rotors on Knife Edges.

may possess. At the same time, few rotors are actually in a state of perfect balance, as no really scientific and accurate method of dynamic balancing has yet been devised for turbines. Thin plates are pinned on as required to those portions of the rotor which are found to be lighter and therefore require added weight.

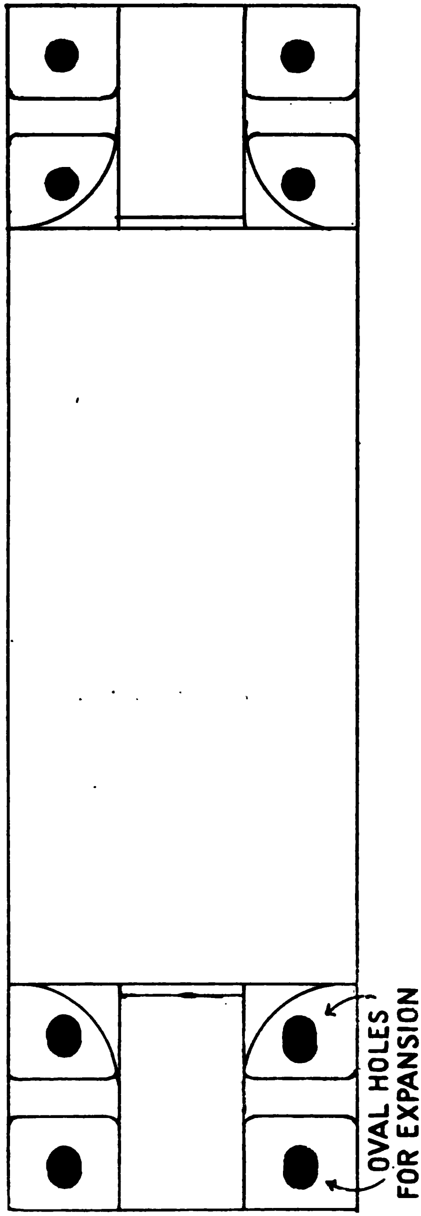
Steaming of Turbines.—The rotors, after undergoing the static balancing tests previously referred to, are placed in the casing and bolted down; the turbine is then

"steamed" for perhaps two or three days. During this steaming process the various clearances are taken by means of "leads," &c., to ascertain the existing conditions when expanded up by heat, and permanent adjustment of the rotor is then made accordingly.

Dummy Rings Adjustment. — When under steam, as described above, the rotor is revolved by steam and screwed up endways at the same time, so that the ahead brass dummy rings make metallic contact with the corresponding grooves, and in this way a bearing surface is obtained between them for all of the rings; any extra hard places on the rings are filed up by hand when the casing is taken off at intervals during the grinding up process; this is done to note what is going on inside. When the rings and grooves are considered to be sufficiently faced up and bearing hard against each other, the rotor is then set aft for, perhaps, $\frac{50}{1000}$ (or $\frac{1}{20}$) of an inch clearance, and a suitable adjustment half ring placed in position in the forward end of the lower half of the thrust block seating to lock the rotor in the required position (see sketch, page 66). It is perhaps unnecessary to state that the dummy clearance varies with the size and power of the turbines. Any longitudinal change of rotor position thus comes first on the dummies, and may result in "seizing up" of the rings and stoppage of the turbine.

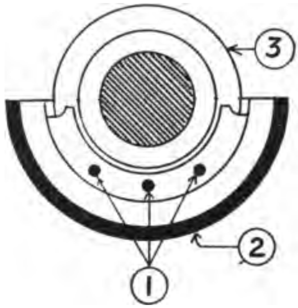
Wear Forward of Rotor.—As the rotor usually wears forward, sometimes the rotor or moving blades are arranged to fit in between the casing or fixed blades, so as to leave rather more clearance between the respective rings on the *forward* side. This allows for wear in the forward direction. (See Clearance Tables, pages 103-105.)

Casing Expansion.—Longitudinal expansion of the rotor casing is allowed for by means of oval bolt holes at the forward end of the seating, this arrangement allowing of slight movement. The after end is bolted down rigidly to the seating of the ship.



Plan of Casing showing Oval Bolt Holes at Forward End for Expansion.

Thrust Blocks.—Small thrust blocks consisting of a large number of brass rings fitting into corresponding collars on the shaft are fitted forward on each turbine. The lower half of each ring is for taking the ahead thrust, and the upper half for the astern thrust. The upper half of the block is adjustable longitudinally by a screwed stud, thus confining the rotor in its correct position fore and aft, a suitable clearance being set of about $\frac{50}{1000}$ of an inch. It is interesting



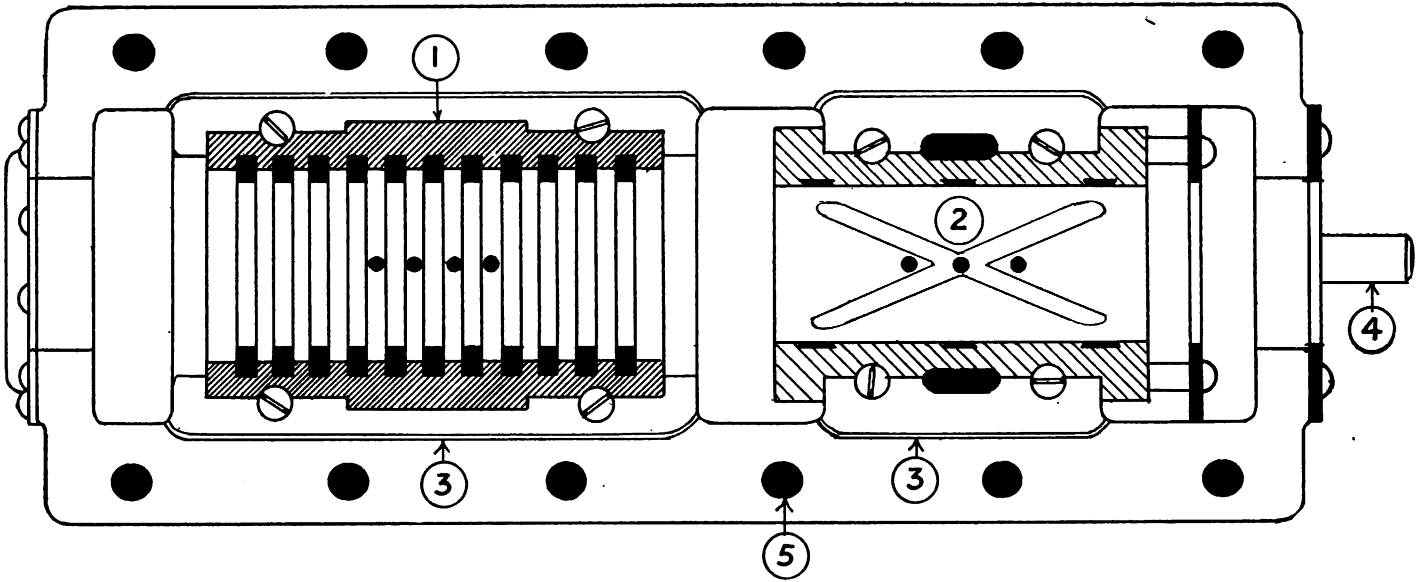
Aft End View of Thrust Block.

1. Oil Holes in Brass Thrust Ring. 2. Adjustment Half Ring. 3. Shaft Collar.

to note that the wear on the block is practically *nil*, as the propeller thrust is balanced by the pressure of steam on the vanes acting in the opposite direction, so that to all intents and purposes the block is not really called upon to receive the thrust as usually understood in connection with ordinary marine engines. The block is chiefly required to take the thrust when steam is turned on or off.

NOTE.—Sometimes the wear takes place *aft* instead of *forward* (usually in the H.P.) owing to the blade steam thrust being in that direction.

End Clearance.—The end clearance of the rotor (measured by thousandths) is adjusted by means of a specially made half ring of brass which fits up hard against the shoulder on the lower half or fixed portion of the thrust block. The longitudinal adjustment of the rotor is thus determined by the position of the lower or ahead portion of the thrust block, and the clearance (about $\frac{50}{1000}$ of an



Inside View of Thrust and Main Bearing Cover.

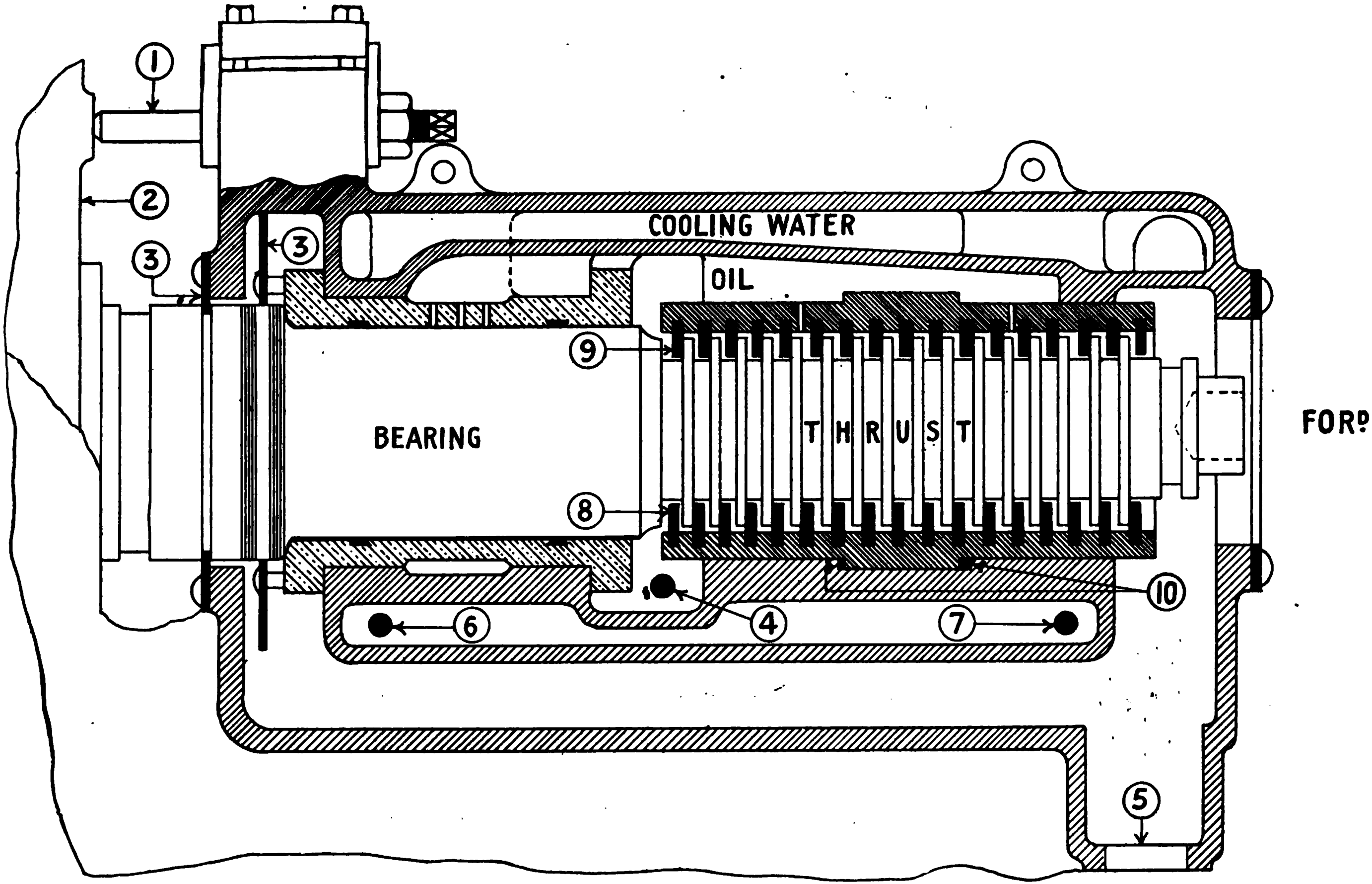
1. Astern Half Thrust. 2. Oil Holes in Brass. 3. Oil Gutters. 4. Adjustment Stud. 5. Oval Holes to allow of adjustment.

inch) can be tested by means of a pin screwed right through the forward end of the casing and adjusted to within a fixed and known clearance of the dummy piston. The longitudinal expansion of the rotor can also be tested by this means.

To adjust Dummy Clearance.—(1) By means of the dogs and "screwing-up" bolt forward, bring the rotor up until the dummy rings and grooves are in actual contact; next measure and note the clearance in thousandths between the after side of the finger plate and the spindle groove when in this position, and screw the rotor back until the clearance between the spindle groove and finger plate is *increased* by say $\frac{30}{1000}$ of an inch, or .03. The lower half or ahead section may then be locked in position by fitting in at the forward end or at both ends the adjusting rings, an allowance of say $\frac{3}{1000}$ clearance being left between the collars and rings for the oil film before doing so.

(2) The upper or astern portion of the thrust is set as follows:—Ease back the nuts of the cover studs, and screw round the adjusting bolt in the cover until the rings and collars come into metallic contact by the cover moving forward. Now screw down tight the cover stud nuts, and adjust the stud until when tested by "feelers" the desired clearance of say $\frac{6}{1000}$, or .006, of an inch exists between the stud and the end of the casing; then ease back the cover nuts, and tap up aft the cover until the point of the adjusting stud and the turbine casing are in actual contact, which will thus give the determined thrust clearance; the cover bolts are then screwed down hard. It will be noted that the total clearance for oil is $\frac{6}{1000}$, or .006. The dummy clearance if when cold is say $\frac{30}{1000}$ of an inch, or .03, usually decreases to $\frac{20}{1000}$, or .02, when expanded after heating up to working conditions. The engineers are therefore instructed to test the dummy clearance with the feeler gauge both previous to heating up and after heating up.

Some chief engineers test and note the clearance referred to every two or three days, or even oftener, this clearance constituting the most important and most delicate adjustment in the whole turbine, and one on which the economy and mechanical efficiency greatly depends. Should the dummy rings and grooves overstep the $\frac{20}{1000}$ or so clearance,

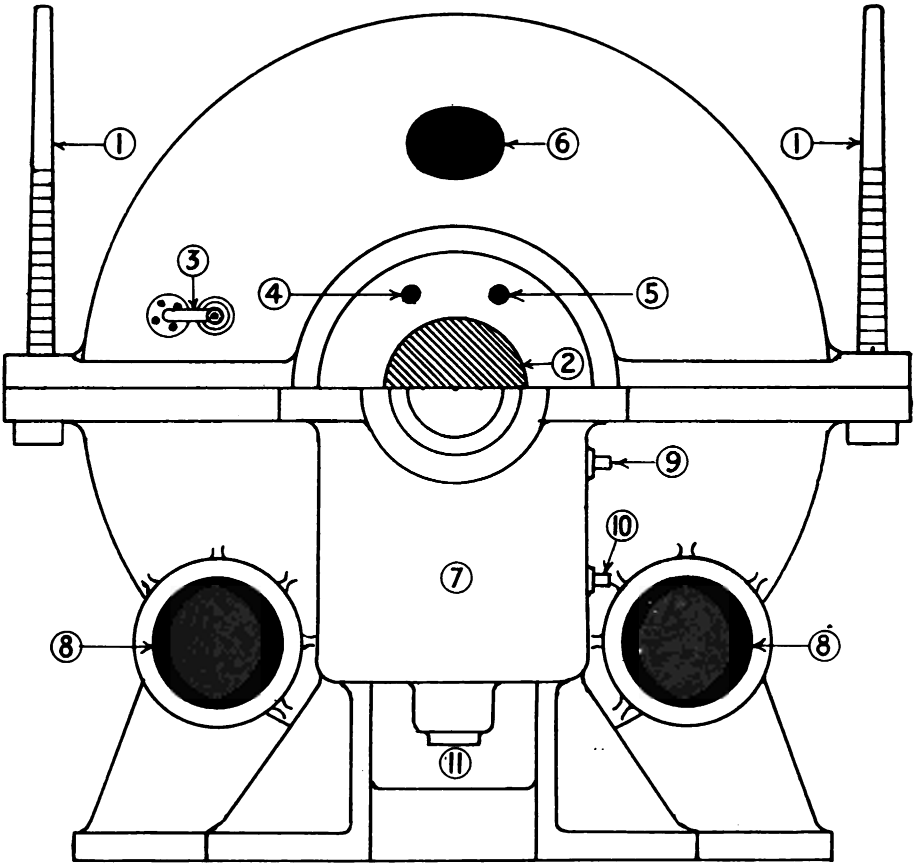


Sectional View of Forward Bearing and Thrust.

1. Adjustment Stud. 2. End of Turbine Casing. 3. Oil Deflecting Rings. 4. Oil Inlet (from oil pumps). 5. Oil Drain to cooling tanks. 6. Cooling Water Inlet. 7. Cooling Water Outlet. 8. Ahead Thrust Half Rings. 9. Astern Thrust Half Rings. 10. Thrust Adjustment Half Rings.

breakdown is sure to occur by the stripping out of the small brass dummy rings ; other damage may also result.

Guide Studs.—Four long guide studs are screwed into the lower half turbine casing, and these are intended to maintain the upper half casing in its correct position



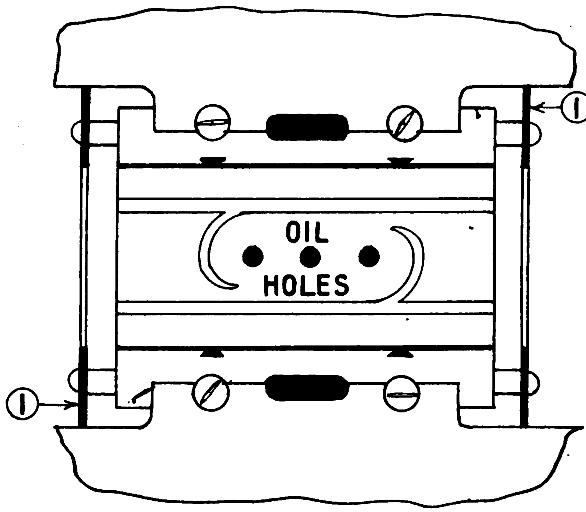
End View of H.P. Turbine (Thrust Cover removed).

1, 1. Guide Studs. 2. Rotor Spindle. 3. Rotor Clearance Gauge. 4. Gland Steam Inlet. 5. Gland Steam "Leak-off." 6. Hand Hole. 7. Thrust Seating. 8, 8. Steam Inlets to Turbine. 9. Oil Supply. 10. Cooling Water Inlet. 11. Oil Outlet.

when being raised or lowered, and thus prevent possible damage to the blades by contact. These studs are placed

two at each end, and they are carefully marked in inches, and parts of inches, to allow of the cover being raised or lowered evenly throughout the length of the turbine, and thus obviate canting.

Main Bearings.—A bearing is fitted at each end of each rotor, and is of the usual marine type, being of brass filled up with white metal. The bearings are held in place by four large screws, each one being partly in the seat and partly in the brass as shown. Small holes are bored through the top or bottom half of the brass, and the oil is forced through the oil holes by the oil pumps, as elsewhere described.



Lower Half Main Bearing.

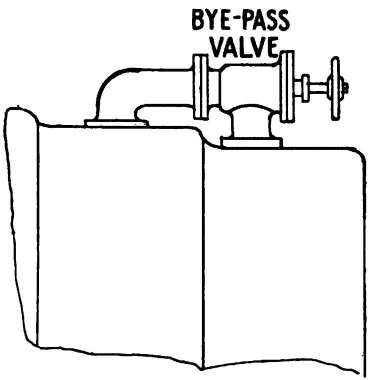
1, 1. Oil Deflecting Rings.

Oil Baffles or Deflectors.—Ring plates of brass in two halves are secured to each end of the after brass, and to the after end of the forward brass, to act as oil baffles, that is, to prevent the ingress of oil to the rotor casing and blades. These baffle ring plates fit in very closely round the spindle circumference and thus limit the oil to the required positions.

Counter Gear.—A worm is fixed to the rotor spindle at the forward end, and by means of suitable worm-wheel and lever transmission gear the revolutions are indicated on a patent speed counter or "tachometer," usually geared to indicate one revolution in ten.

The main bearing brasses are fitted with a few oil holes at the top or bottom, through which the oil is forced upwards into the bearing by means of the pressure at which the oil pumps work—about 8 lbs. per inch.

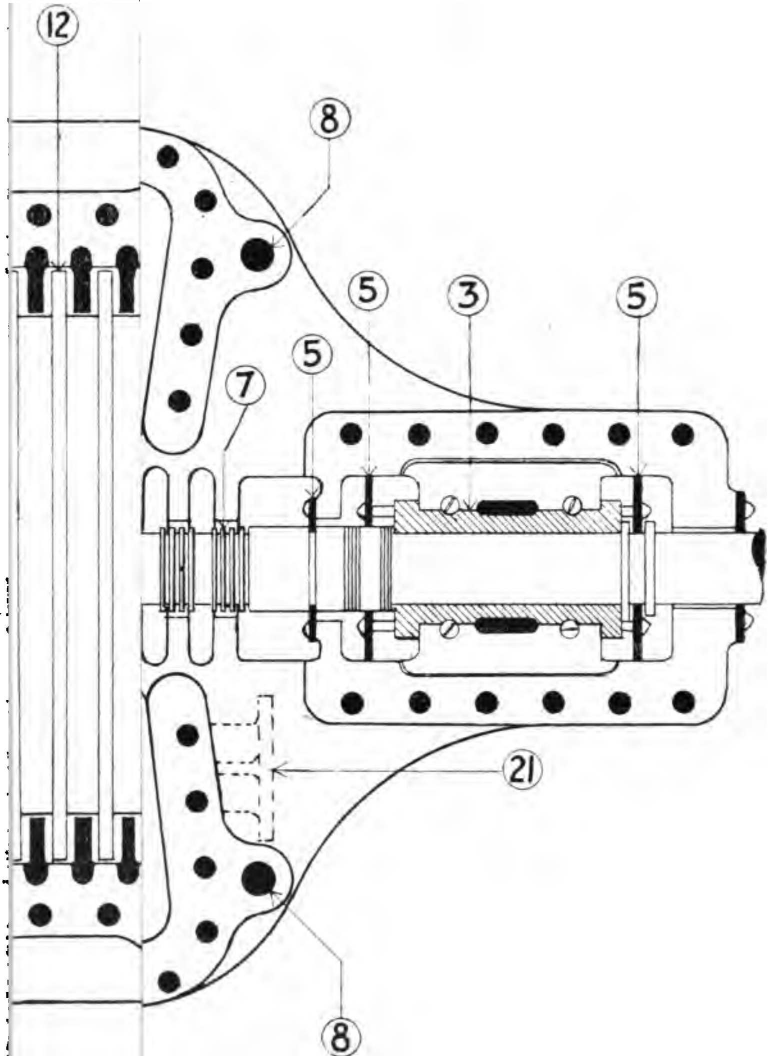
Bye-Pass Valve.—Steam from the first expansion can be admitted direct to the second or third expansion by means of a bye-pass valve if required. This is usually



Bye-pass from First to Third Expansion (on H.P. Turbine only).

done in starting the turbine, and assists in reducing excessive shock on the first rings of blades. It is also employed in developing increased power and speed when required. This connection is only fitted to the H.P. turbine.

Rotor Screw.—When the rotor requires to be moved longitudinally, a pin is screwed into the forward end of the spindle, which is tapped out for the purpose, and by means of a dog arrangement and double nuts the rotor can be moved forward or aft as required. It is sometimes necessary to disconnect the first length of shafting when screwing up the rotor as described.



Lower E

"stoppers"

ger" plate. (12) Second expansion. (13) Third expansion.

expansion.

[*To face page* 72.

Turning Gear.—A gear wheel is keyed to the rotor spindle at the joint formed by the first length of shafting just outside the after end of the casing, and by means of ordinary turning gear the rotor can be moved round as required. This may be necessary when taking the various blade and casing clearances, or when making examination of the rotor or casing.

Strainer.—Each steam pipe has a brass wire strainer fitted inside a special chest bolted on the casing to prevent the possibility of dirt of any kind entering the turbines; it also acts, to some extent, in checking priming.

Steam Connections.

The usual steam connections on the rotor casing are as follows:—

Main or Ahead Turbines.

1. **Steam Inlet at Forward End.**—In the H.P. turbine this is direct from the boilers, and in the L.P. turbines from the H.P. exhaust.

2. **Exhaust at After End.**—In the H.P. turbine this leads to the two L.P. turbines as initial steam, and in the L.P. turbines direct to the condenser.

3. **L.P. Turbine Non-return Valves.**—These valves, arranged to prevent the return of steam from the L.P. to the H.P. turbines, are fitted in a chest on the L.P. casing where the H.P. exhaust branch connects, and being loaded by means of two springs, only allow of the admission of steam from H.P. to L.P. casings, but close immediately steam attempts to pass back to the H.P. from the L.P. These valves come into action when high-pressure steam is supplied direct to the ahead L.P. turbines in manœuvring or working with the outside shafts only.

4. **Direct Steam to L.P. Turbine.**—This supplies high-pressure steam direct to the ahead L.P. turbines, and is required when running ahead with the outside shafts only.

5. **Bye-Pass Steam.**—This consists of a hand-valve and pipe connection leading from the first to the third expan-

sion to admit high-pressure steam direct to the third expansion if required to increase the power, or to create an equalising pressure in starting up and prevent possible damage to the blades by vibration.

6. **Escape or Relief Valves.**—Escape valves, loaded to a suitable pressure, are fitted on the top of the two L.P. turbine casings forward, and one on the H.P. turbine casing aft.

7. **Gland Steam and "Leak-off."**—High-pressure steam is supplied to the H.P. glands by a small pipe and cock and lower-pressure steam to the L.P. turbines. In addition to the steam inlet the H.P. turbine is fitted with a "leak-off" from the second series of rings and collars to the L.P. turbine, or to the condenser. The L.P. gland leaks off to the third expansion, or more generally, to the condenser. This assists in preventing steam leakage outwards.

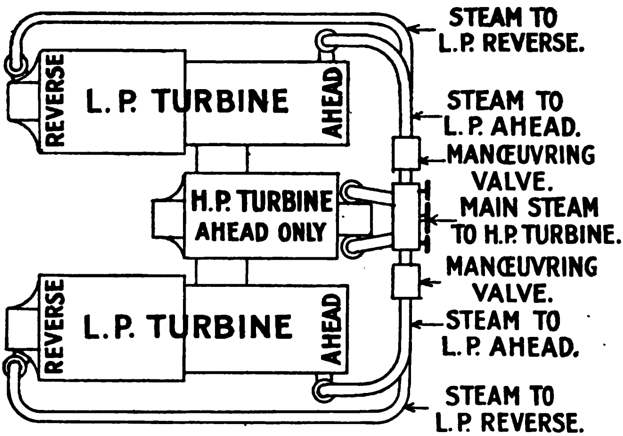
8. **Dummy "Leak-off."**—Two pipes are usually fitted from the ahead dummy casings to the third expansion of the turbine, where the pressure is less, to lead away any steam which may leak through the first series of dummy rings and grooves.

Reverse Turbine.

9. **Direct Steam.**—The reverse turbines only require one connection, that of the direct steam at the after end, which, after expanding through the reverse turbine blades forwards, exhausts into the condenser by the common exhaust branch. Small drain holes are fitted in the bottom of the reverse turbine casing.

Steam and Reverse Valves. — Steam can be admitted by the large hand-controlled valve to the H.P. turbine direct, and so through the series of turbines, so that all are working at once. Two smaller hand valves also admit full-pressure steam to a pair of chests, each containing a piston valve which is actuated by patent steam and hydraulic gear. Round ports are cast in the chest, top and bottom, one end admitting steam to the ahead L.P. turbine, and the other admitting steam to the reverse L.P. turbine, and the valve is moved up and down by

the gear to uncover the ports and admit steam from the centre of the piston valve as required, the steam at the



Steam and Reverse Valve Arrangement.

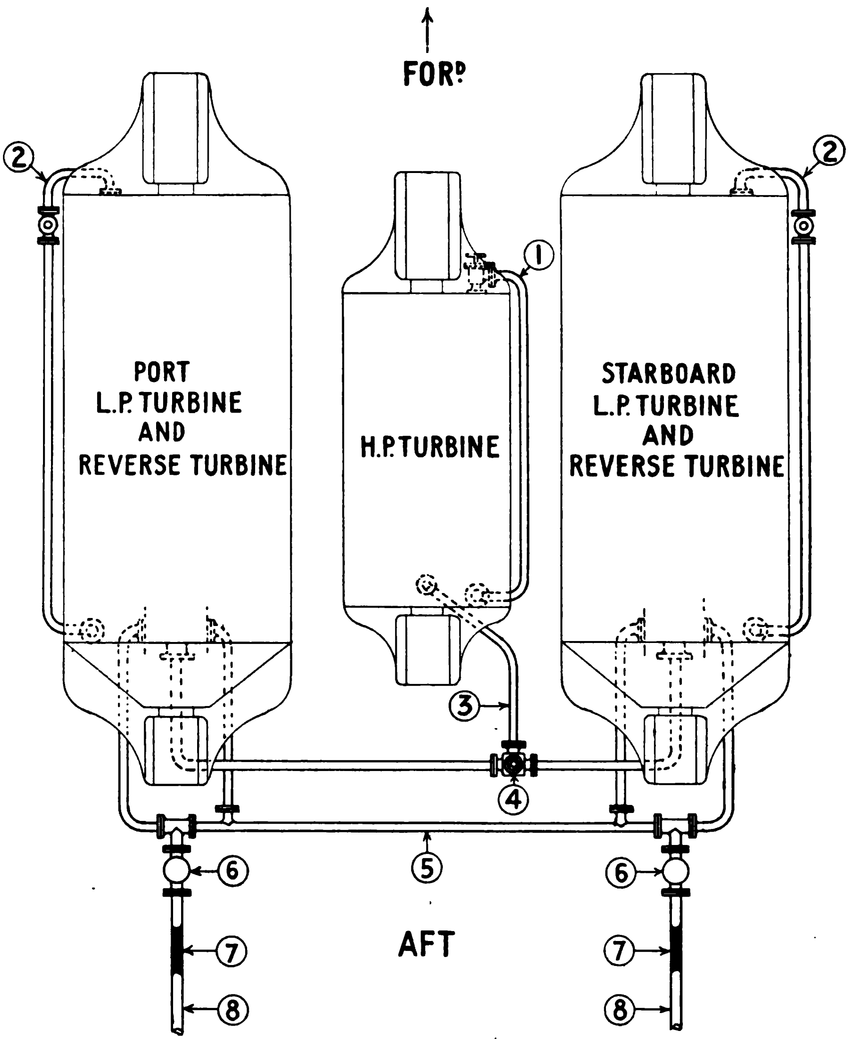
same time being admitted to the chest itself by means of the hand-valve referred to.

This constitutes the reversing and manœuvring gear.

Turbine Casing Drains (Forward).—A drain pipe and cock or screwdown valve connects the forward end of each turbine casing with one of the last expansions aft. This allows any condensed water to drain out from the higher forward position of the casing to the lower after position. These drain connections are only kept open when the turbines are stopped temporarily.

NOTE.—This connection is not always fitted.

Turbine Drains (Aft)—*L.P. Turbines.*—Large double drain pipes and specially light non-return valves are fitted at the after end of each L.P. turbine, and these are connected to the "wet" or ordinary air pumps which are kept running while the turbines are stopped temporarily. This ensures the withdrawal of any condensed water from the turbine casings. To prevent air finding its way back into the L.P. turbine casings, a **U** bend is arranged on the air pump suction pipe to act as a "water seal," and as this



Turbine Drain Connections.

1. Drain from forward to after end of H.P. Turbine.
2. Drain from forward to after end of L.P. Turbines.
3. Drain from after end of H.P. Turbine to after end of either L.P. Turbine.
4. Two-way Cock for changing over.
5. Drain from after end of both L.P. Turbines to "Wet" Air Pump.
6. Light Non-return Valves.
7. U Bend in Pipe to act as "Water Seal" and prevent return of Air.
8. "Wet" Air Pump Suction Pipes.

NOTE.—The connections marked 1, 2, and 3 are only open when the turbines are stopped, and the connections marked 5, 6, 7, and 8 are open when the turbines are either running or stopped (temporarily).

bend contains water, the admission of air is prevented even should the air pumps cease working. The non-return valve mentioned above would prevent the possible return of water back from the pump into the turbine casings.

H.P. Turbine.—The drain from the after end of the H.P. turbine is merely led to the drain pipe connection of either of the L.P. turbines, a cross or "change over" connection to either L.P. being arranged by means of a two-way cock.

NOTE.—The water which is constantly drained off from the L.P. turbines when running is formed by the adiabatic expansion of the steam in the turbines. The H.P. exhaust pipes to each L.P. turbine are placed low down to allow the water condensed in the H.P. turbine to pass easily to the L.P. turbines, and afterwards be drained off by the connections aft to the "wet" air pumps. (See sketch facing page 74.)

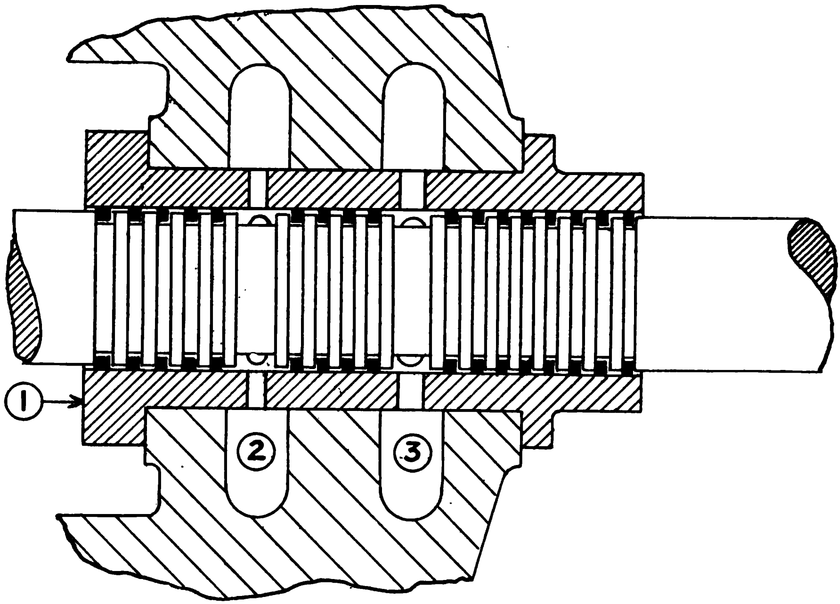
All casing drains can be opened up to the bilges when required, and this is done when the engines are rung off, so as to get rid of the vacuum existing in the turbine casings.

The two drains led from the H.P. exhaust pipes to the L.P. casings aft are just kept eased off the face when running, so as to allow the condensed water only to be drained off.

NOTE.—Dirty boiler water may be said to be fatal to turbines, and if feed filters are necessary fittings with reciprocating engines they are indispensable with turbines, as, even with the strainers fitted on the turbine ends of the steam pipes, any grit or dirt present finds its way into the turbine casings, and serious disablement may result. Dirty water, it should perhaps be mentioned, produces priming in the boilers, hence the passage of the foreign matter into the steam pipes and turbines.

Steam Glands.—The "steam glands" consist of a cast-iron case, a series of small collars on the shaft, and a corresponding set of loose fitting cut-brass rings fitting in between the collars and bearing outwardly on the case. Steam is admitted by a cock through ports to the inner

surface of the case and rings, and by this means any leakage outwards from the rotor casing is prevented, and



Steam Glands.

1. Case.
2. Outer Pocket (Steam Inlet).
3. Inner Pocket (Leak-off).

NOTE.—The outer pocket usually carries a pressure of 1 or 2 lbs., while the inner pocket or leak-off shows a vacuum of about 15 inches or thereabout.

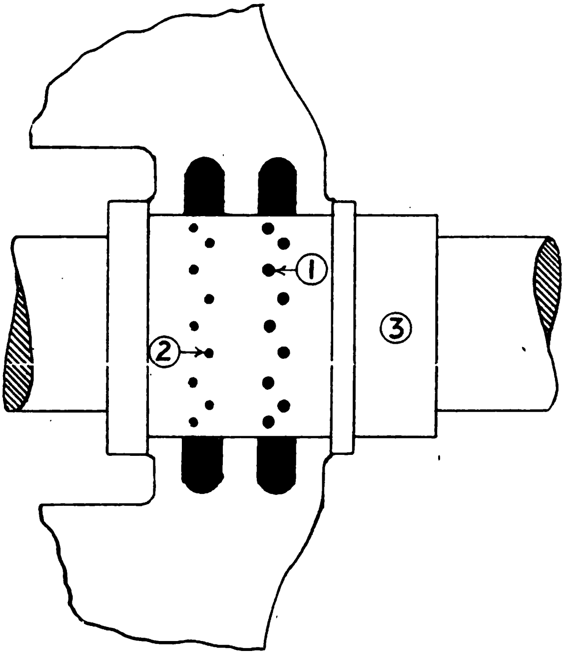
The small brass-cut rings are often found to revolve with the shaft, although not intended to do so. This results in grooving of the case and wear of the rings.

Observe how the rings are bearing on the shaft collars sideways.

in the case of the L.P. any leakage of air inwards is also prevented. A steam gland is fitted to each end of each turbine.

Steam Gland Inlet (Outer Pocket).—Steam is admitted by a pipe and cock to a port or "outer pocket," and thence to the outside of the set of holes in the gland case, and passing to the inside enters the gland rings and collars, and thus prevents outward leakage. The steam pressure gauge usually indicates about 1 lb. pressure.

Steam Gland "Leak-Off" (Inner Pocket).—Another connection by means of small holes is made between the inside of the gland and a port on the outside, called the "inner pocket," and any steam that leaks past the second (inward) set of collars and brass rings next the rotor passes outwards and "leaks off" by means of a cock and pipe connection to the third expansion of the L.P. turbines, where a vacuum exists. The compound gauge



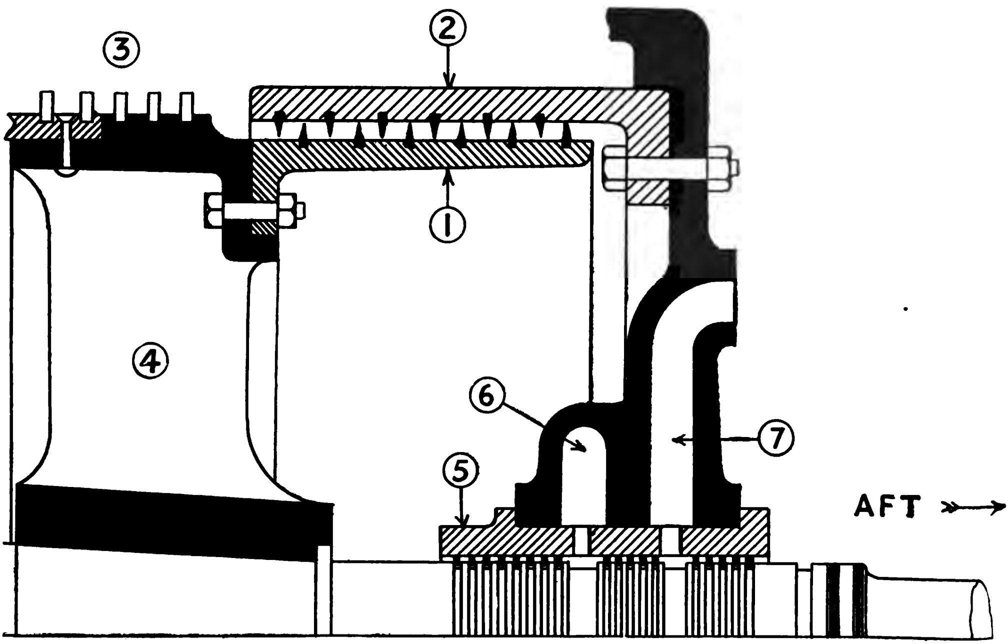
Steam Gland Casing in Position.

1. "Leak-off." 2. Steam Inlet. 3. Gland Case.

connected to the inner pocket usually indicates a vacuum of about 15 inches or thereabout. The steam glands of the H.P. turbine have to resist a pressure of about 25 lbs. or so, due to the H.P. final or exhaust pressure flowing into the inside of the drum, whereas the L.P. steam glands have only to keep out the atmospheric pressure, as there is a vacuum in the inside of the drum (L.P. exhaust).

In both the H.P. and L.P. turbines the exhaust steam from

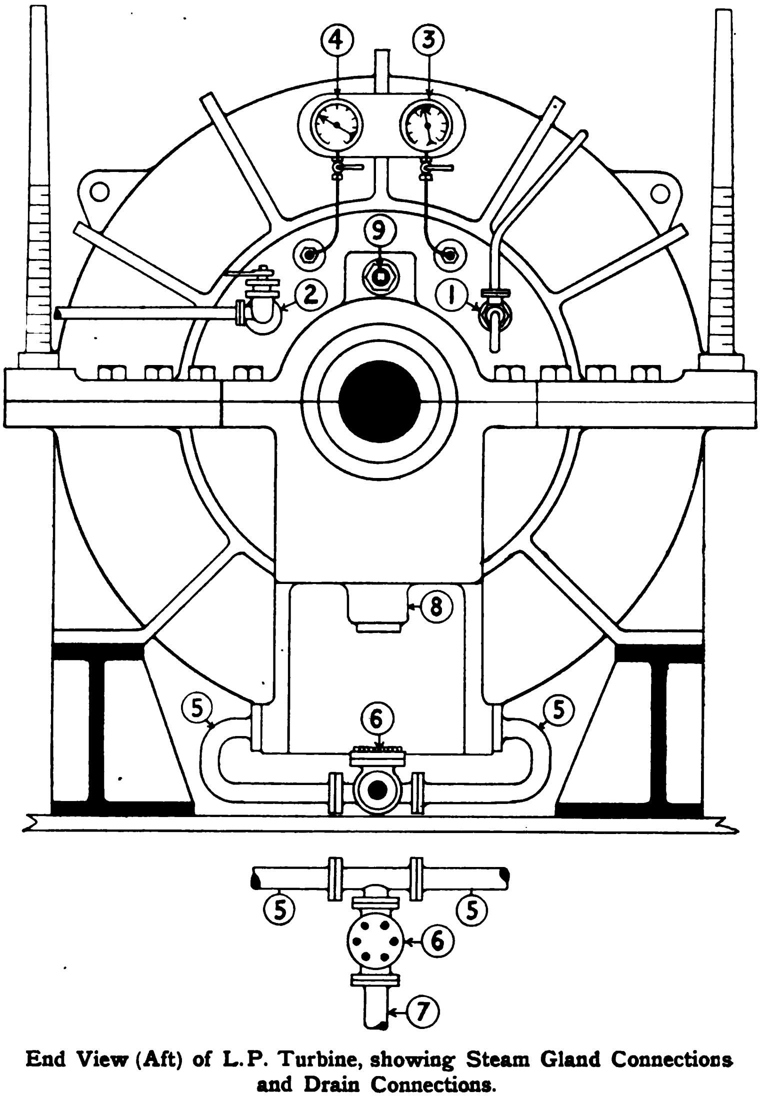
the last ring of blades flows into the *inside* of the drums, and the steam glands are therefore required to pack the spindle against outflow of steam in the case of the H.P. turbine, and against the admission of air in the case of the L.P. and reverse turbines. The pressure inside of the H.P. rotor drum is usually somewhat between 10 and 25 lbs., and the vacuum inside the L.P. rotor drum approximates to that carried in the condenser, being probably between 2 or 3 lbs. less. As before stated, the H.P., L.P., and reverse dummies act as packing to prevent the outflow of the higher or admission pressure steam from the turbine casing into the atmosphere; whereas the steam glands on the rotor spindles are only required to pack the turbine against the exhaust or lower pressure steam.



Section View of After End of L.P. Turbine.

1. Reverse Dummy Piston.
2. Reverse Dummy Casing.
3. First Reverse Expansion.
4. "Wheel" or "Centre."
5. Steam Gland Case.
6. Inner Pocket ("Leak-off").
7. Outer Pocket (Steam Inlet).

NOTE.—The "fins" of the reverse dummy are usually pitched about 1 inch apart, and are formed of ⅛-inch brass.

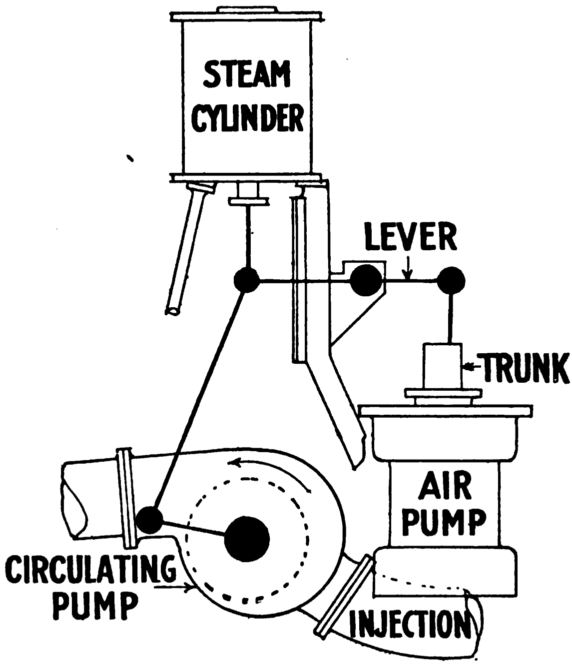


End View (Aft) of L.P. Turbine, showing Steam Gland Connections and Drain Connections.

1. Steam Inlet Cock admitting Steam to Outer Pocket.
2. Leak-off Cock on Inner Pocket connecting to 3rd or 4th L.P. Expansion.
3. Gauge on Outer Pocket.
4. Gauge on Inner Pocket.
5. Drains from Turbine Casing Bottom.
6. Light Non-return Valve.
7. Suction to "Wet" Air Pump.
8. Oil Drain to Oil Cooling Tanks from Thrust and Bearing.

Pumps.—All pumps fitted are of the independent double-acting type, and are as follows :—

Air Pumps.—The most recent practice consists of the fitting of independent twin air pumps of the Weir type, and technically known as "wet" air pumps. These pumps draw as usual from the bottom of the condenser, and deliver the water into the feed tank and gravity feed filter connections; from these the water is pumped by other independent donkey pumps into the surface feed heater. and float tanks overhead, and from there the water is taken by the feed pump proper and delivered direct into the boilers, purified and heated.

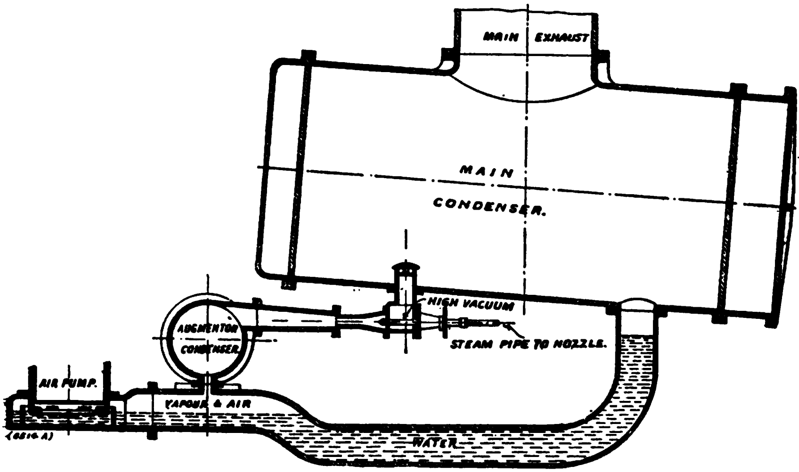


Air and Circulating Pumps.

"Dry" Air Pump.—To obtain good efficiency in turbines, a high vacuum is of the utmost importance, and to produce this a special pump, known as a "dry air pump," has been recently introduced by Messrs G. & J. Weir, in addition to the ordinary independent air pumps as formerly fitted.

The Weir "dry air pump" draws the air or vapour only from the condenser, and the ordinary air pump draws away the vapour still left and the condensed water. The dry air pump is therefore placed high up in position (usually above the circulating pump engine) with the suction branch from the condenser and the discharge pipe led away over the ship's side. By the aid of this special pump the vacuum carried has been as high as 29 inches. The dry air pump chamber is kept cool by means of a water jacket. Regarding the subject of vacuum the Hon. A. C. Parsons says: "An addition of 1 in. to the vacuum in the condenser over 26 in. deducted 4 per cent. from the steam used, a further increase of the same amount 1 in. meant a further gain of 4½ per cent., while 29 in. brought the steam consumption down 5½ per cent. more."

Another arrangement, called a "vacuum augmenter," devised by the Parsons company, has been applied to increase the vacuum above that obtained by the air pump, and, as shown in the sketch, consists of a smaller auxiliary

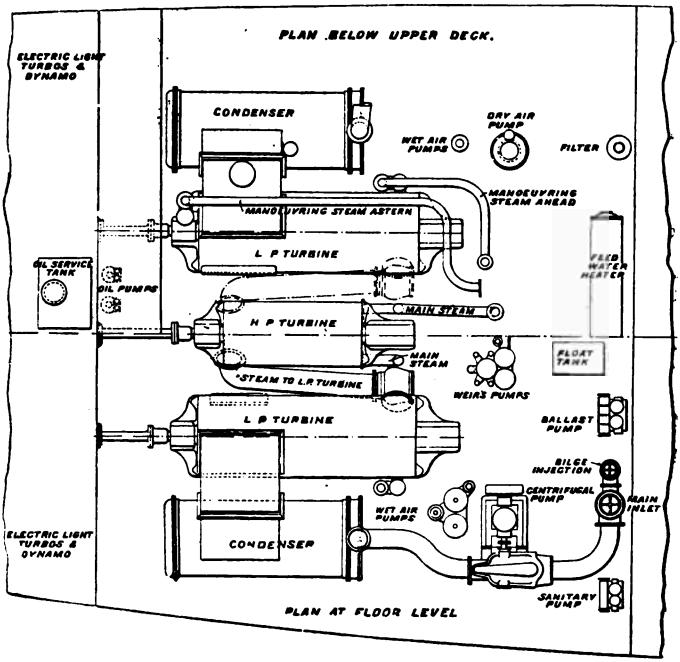


Vacuum Augmenter.

condenser placed below the main condenser and connected to it by a pipe having a conical contracted portion through which a jet of steam is forced. The effect of this is to exhaust most of the air and vapour from the condenser, and deliver it to the air pump. By arranging the air pump

suction pipe with a dip as shown, the air and vapour is prevented from returning to the condenser by the water contained in the pipe, thus forming what may be termed a "water-seal." The "vacuum augmenter" has been found in some cases to increase the vacuum by about 2 inches above that obtained with the air pump.

It should, however, be stated that the "Weir" dry air pump has been fitted to all the most recently completed turbine steamers.



Plan of Turbine Room.

Circulating Pumps.—These two pumps, one for each condenser, force the cold sea water through the condenser tubes as in ordinary practice, and assist, in combination with the air pump, in producing condensation of the steam and the formation of a vacuum.

Feed Pumps.—As before stated, these pumps, arranged in pairs, are usually of the "Weir" or some other well-known patent type.

TURBINES OF S.S. "QUEEN." (Channel Steamer.)

Various Pumps.—Other sets of pumps are required to circulate the cooling water through the bearings and cooling tanks, to force the oil at a pressure of 8 or 10 lbs. per square inch through the bearings, and for numerous other services, such as bilges, ballast tanks, &c. It will thus be apparent that the number of independent service pumps required is in excess of those commonly fitted in engine-rooms of the reciprocating type.

Revolutions.—As previously stated, high revolution speed is necessary for the economical running of turbine engines. In the Clyde turbine steamers the L.P. shafts revolve at about 800 revolutions per minute, and the H.P. shaft from 550 to 650 revolutions per minute, but the revolutions vary considerably with the type of vessel and speed desired. In ocean-going steamers the revolutions are much less than the above, the "Carmania" being only 185 per minute. This means a corresponding increase in the size of the rotor, and therefore increased weight of machinery. Recently the speed of all three shafts has been arranged to run at very nearly the same number of revolutions per minute. In the three large Irish Channel steamers, "St George," "St David," and "St Patrick," recently completed for the Great Western Railway Company, the revolutions run were just under 500 per minute at full power (10,000 I.H.P.), for a trial trip speed of 23 knots.

Governors.—Governors are now rarely fitted to turbines, but the "Carmania," it may be stated, is supplied with "Aspinall" type governors.

Speed Indicator.—The revolutions are indicated by an appliance known as a "Speed Indicator," or "Tachometer," which is fitted with a figured dial and pointer, and is specially employed in registering high revolution speeds in dynamos, &c. This instrument registers 1 revolution in 10.

Indicated Horse-Power.—The indicated horse-power of a turbine engine cannot be calculated by any

rule at present formulated, but can only be approximated by tank experiment or by comparison with motors or engines of known power. The "King Edward" (river steamer) is estimated as being of 3,500 indicated horse-power, and the "Queen" (cross-Channel steamer) 8,000 I.H.P., while the "Carmania" is estimated as being equal to 21,000 I.H.P., and the "Express" Cunarders 65,000 I.H.P.

The *shaft* horse-power can, however, be calculated by means of the "Torsion meter" (see page 141).

Boilers.—Ordinary "Scotch" (either single or double-ended) boilers are used, working at a pressure of from 150 to 200 lbs. per square inch. The "Carmania" is supplied with thirteen "Scotch" boilers, working at a pressure of 195 lbs. per square inch, and the new "Express" Cunarder steamers have twenty-five boilers.

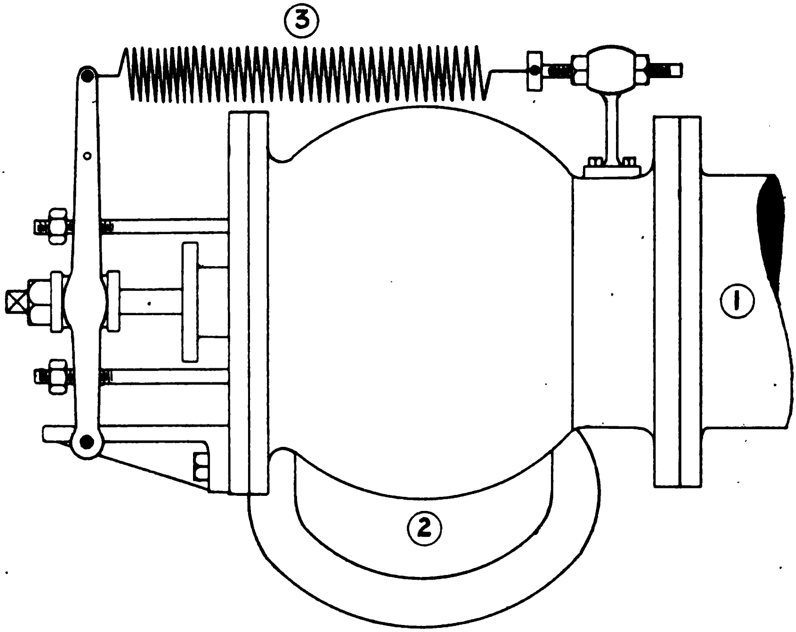
Working of Turbines.

Going Ahead.—In going ahead full speed, steam is admitted by the main steam valve to the H.P. turbine, which it enters, and passing through the rings of guide and shaft blades alternately and rotating the shaft, exhausts at the other (after) end to the two L.P. turbines, one on each side.

Entering the L.P. turbine casings, the steam passes through the various rings of blades, and finally exhausts at a very low pressure into the condensers. The independent air and circulating pumps maintain the vacuum, and Weir's patent feed-pumps deliver the feed-water into the boilers after it has passed through the feed-heater and feed-filter as commonly arranged.

Going Astern.—In going astern the main steam valve is closed and full-pressure steam is admitted by two valves to the astern turbines placed aft in the L.P. casings. When this is done, two large non-return valves, each fitted with a spring and placed between the H.P. and L.P. turbines, close automatically and prevent the return of steam from the L.P. turbine back into the H.P. turbine.

The H.P. turbine is then in a vacuum, and the shaft and blades revolve idly as the propeller is set in motion by the water acting on the blades. Very little power is absorbed in this, as the vacuum offers but slight resistance to the rotation of the H.P. turbine blades.



Spring-Loaded Non-Return Valve.

1. Exhaust from H.P. Turbine. 2. Admission to L.P. Turbine.
3. Pair of Springs.

This valve is fitted on the L.P. end of the H.P. exhaust pipes, and is intended to prevent the return of steam to the H.P. turbine when running with the L.P. turbines only.

Manœuvring.—As before stated full-pressure steam can be admitted to the two L.P. turbines to drive ahead, the main steam being closed, or can be admitted to the two reverse turbines on the L.P. shaft to drive astern, the main steam valve still being closed, and the H.P. turbine revolving easily in a vacuum. It is also possible to give steam to one L.P. ahead turbine and the other L.P. reverse turbine at the same time, to allow of quick turning of the vessel or for manœuvring at piers, &c.

It will be obvious from the above that when going ahead with the two L.P. turbines, the L.P. reverse turbines are then running in a vacuum, and when going astern with the L.P. reverse turbines the L.P. ahead turbines are running in a vacuum.

Reversing. — Contrary to the first impressions and general expectations, the turbine steamers have proved that quick stopping and reversing can easily be accomplished if the reverse turbines are made of sufficient power. The "Dieppe" (cross-Channel steamer) has supplied record figures for quick stopping and going astern, so that further doubt on this point is out of the question.

It should be stated that sudden reversing is very severe on the turbine, as excessive vibrations are set up by the change of rotation.

Speed Regulation.—The engineer in charge regulates the speed as indexed by the telegraph, entirely by means of the revolution counters and the pressures shown on the various gauges and dials in connection with the turbine casings. Gauges are connected to the following points:—

1. Main steam pipe.
2. H.P. turbine.
3. Port L.P. turbine.
4. Starboard L.P. turbine.
5. Port L.P. astern turbine.
6. Starboard L.P. astern turbine.
7. Starboard condenser.
8. Port condenser.

NOTE.—The gauges from No. 2 to No. 8 are of the "compound" type, and indicate either pressure or vacuum as required, according to the working of the various turbines.

Other gauges are fitted to show the oil pump pressure, cooling water pump pressure, &c., &c.

Oil Supply.—Small oil pumps are used to force oil under a pressure of about 8 lbs. per square inch through

holes in the white metal bearings fitted at each end of the turbines. The oil is taken from a tank, and after making the circuit of the bearings, is returned again to the tank coils to be cooled and used over again. Below the oil service passage in the bearing block there is a cooling water service passage through which a stream of cold water is forced at a pressure of about 12 lbs. by a special pump.

It is of the utmost importance that the oil service be absolutely regular and uninterrupted, as if not, complete disablement of the turbine may result.

The regular maintenance of the oil service is of the highest importance for the successful running of the turbines, as should the oil supply cease, rapid heating up and melting out of the main bearings would result. This, again, by the subsequent wear-down, might produce stripping of the vanes of the rotor and drum. A sight glass, through which the oil passes, is often fitted, or a small cock with a hole bored out through the shell so that the oil spurts out when the cock is turned.

It is advisable, however, not to use the same oil for too long a period, as oil repeatedly used has its lubricating properties somewhat impaired, so that fresh supplies of unused oil become necessary.

Difficulty is occasionally found in keeping down the temperature of the oil supply, which, if once heated up, has a decided tendency to remain very warm, the cooling water having little apparent effect in lowering the temperature.

NOTE.—The specific heat of oil is .36.

Oil is only used in the bearings at each end of the turbines, and is prevented from entering the turbine casing by the oil baffle rings or "deflectors." The condensed steam, therefore, contains less oily matter than usual, with the resulting advantages to the boiler of less danger of collapsed furnaces and less pitting and corrosion.

Shaft Rotation Indicator.—Sometimes the forward end of the thrust block casing is fitted with a thick glass window and a point attached to the centre of the rotor spindle end, so that the direction of shaft rotation

can be observed from the outside; arrows indicating the "ahead" and "astern" direction of shaft rotation are also marked up on the casing end, which therefore leave no room for doubt as to whether any particular turbine is actually running ahead or astern.

Wear.—The wear on the brass blades of the turbines is practically *nil* after some years of service, the friction of the steam apparently having little or no effect on them. Wear on the bearings is very slight indeed, and only shows when something gets out of line. It is understood that when running the turbines balance themselves all round, or in other words are floating more or less, thus minimising the wear to a great extent.

As mentioned previously, very little pressure bears on the thrust block, as the blades and dummy pistons together take up and counterbalance most of the propeller thrust.

It is also evident that gyrostatic action of the rotors is so slight as to be negligible altogether. "Whipping" is also eliminated owing to the very careful balancing to which the various parts are subjected while under construction.

Wear of Gland Rings.—The loose brass rings which constitute the steam gland packing occasionally show decided signs of wear after several months' service, and if not renewed some of the rings are apt to give way altogether. A broken ring might thus result in damage to the blades, as the broken pieces could pass into the rotor drum and from there out again into the case and among the rings of blades. This is quite possible, as the wheels or centre have eight large openings cast in them which open up the drum at each end to the exhaust pressure of the turbine, and therefore to the casing.

If one or two of the H.P. turbine gland rings wore thin and finally broke, the pieces might pass into the drum as described, and then out of the drum and into the L.P. turbine with the exhaust steam, and damage to the L.P. blades would be likely to follow.

From the foregoing it will be obvious that the gland rings should be regularly renewed and kept up to their proper size and strength.

Wear of Thrust Block.—After long service the thrust block rings and collars show practically no wear, the surfaces being merely polished, and in many cases the original tool marks can still be made out. The wear, measured by the writer, in one case was only $\frac{2}{1000}$ of an inch after two years' hard running.

Wear of Bearings.—The wear of the bearings is also very slight indeed, and in some cases is a negligible quantity, being merely $\frac{2}{1000}$ or $\frac{3}{1000}$ either *down or up*, as sometimes the bearings appear to wear about equally all round, which circumstance would indicate that the rotor when running is what is called "floating." The wear down of the L.P. rotors in a case tested by the writer was only $\frac{4}{1000}$ of an inch after two years' service.

Wear of Blades.—As a general rule the turbine blades after fairly long service show no sign of wear whatever, the only difference noticed being the darkened colour produced by the effect of the heat of the steam, otherwise the blades are unchanged, the binding and brazing being as originally finished.

Vibration.—Absence of machinery vibration is a point of considerable importance, especially in the case of a passenger steamer, and this advantage may be fairly laid claim to for turbine engines, the machinery vibration being practically *nil*, although the propellers often set up severe vibrations aft just over the stern.

R.M.S. "Viper."—This express turbine steamer, which is intended for the daylight service of Messrs G. & J. Burns Limited, between Ardrossan and Belfast, is 325 feet in length, 39 feet 6 inches in breadth, 24 feet moulded depth to spar deck, and a gross tonnage of 1,750 tons.

Built to the highest class of Lloyds' special survey and to the Board of Trade requirements, she represents the accumulated experience of the Fairfield Works, so far as structural details are concerned. The main propelling machinery consists of Parsons turbines; the guaranteed speed at sea is 21 knots, so that the passage between Ardrossan and Belfast should only occupy $3\frac{3}{4}$ hours.

The propelling machinery consists of three compound turbines, one high pressure and two low pressure; each of these turbines drives a separate shaft. The high-pressure turbine is on the centre shaft and the low-pressure turbines on the outer or wing shafts; with the latter are incorporated the reversing turbines which work in vacuum when the ship is going ahead; the reversing turbines can stop the vessel when going at full steam ahead in one and a half minutes from the time that the engineer receives the order from the captain. The turbines are of the well-known Parsons type, and made by the builders under license from the patentee.

The vessel has four double-ended boilers 20 feet 6 inches long and 14 feet in diameter, constructed for a working pressure of 165 lbs. per square inch. The air pumps and boiler feed-pumps are supplied by Messrs G. & J. Weir Limited. The bunkers have a capacity for 120 tons of coal.

NOTE.—On the trial runs over the measured mile the "Viper" attained a speed of fully 22 knots, with an air pressure of only ½ inch water, while on the Cloch to Cumbrae and back continuous run the mean speed was fully 21 knots, with natural draught and easy steam.

Advantages.

Some of the advantages of the turbine over the ordinary reciprocating engine may be stated as follows:—

1. Fewer working parts, as no piston or slide valves, piston rods, &c. &c., are required.

2. Steam applied direct from the boiler to the shaft without any intervening loss of power.

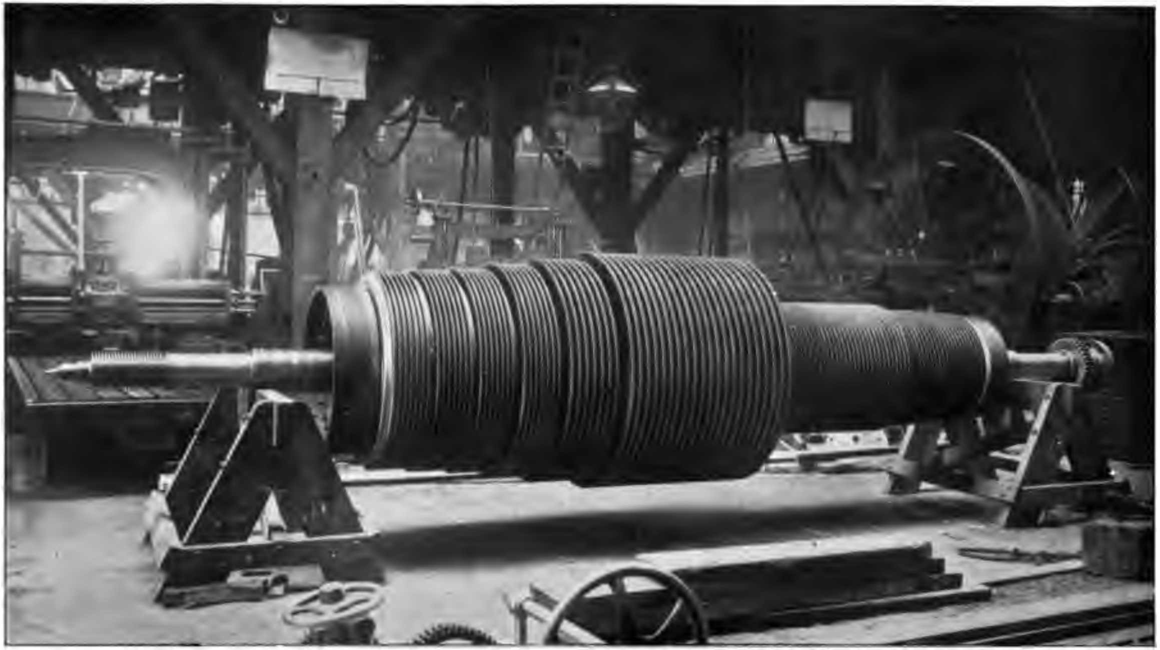
3. Less danger of breakdown, as it is almost impossible for the small vanes to become damaged, unless actual contact takes place between the blades and casing, or between the rings of rotor and casing blades. In addition to this, three separate lines of shafting are provided, so that if one became disabled, two are still left for running with.

4. Less weight of machinery for the same power.

5. Engines placed well down in the vessel.

6. Greater speed for the same consumption of coal at high speeds.

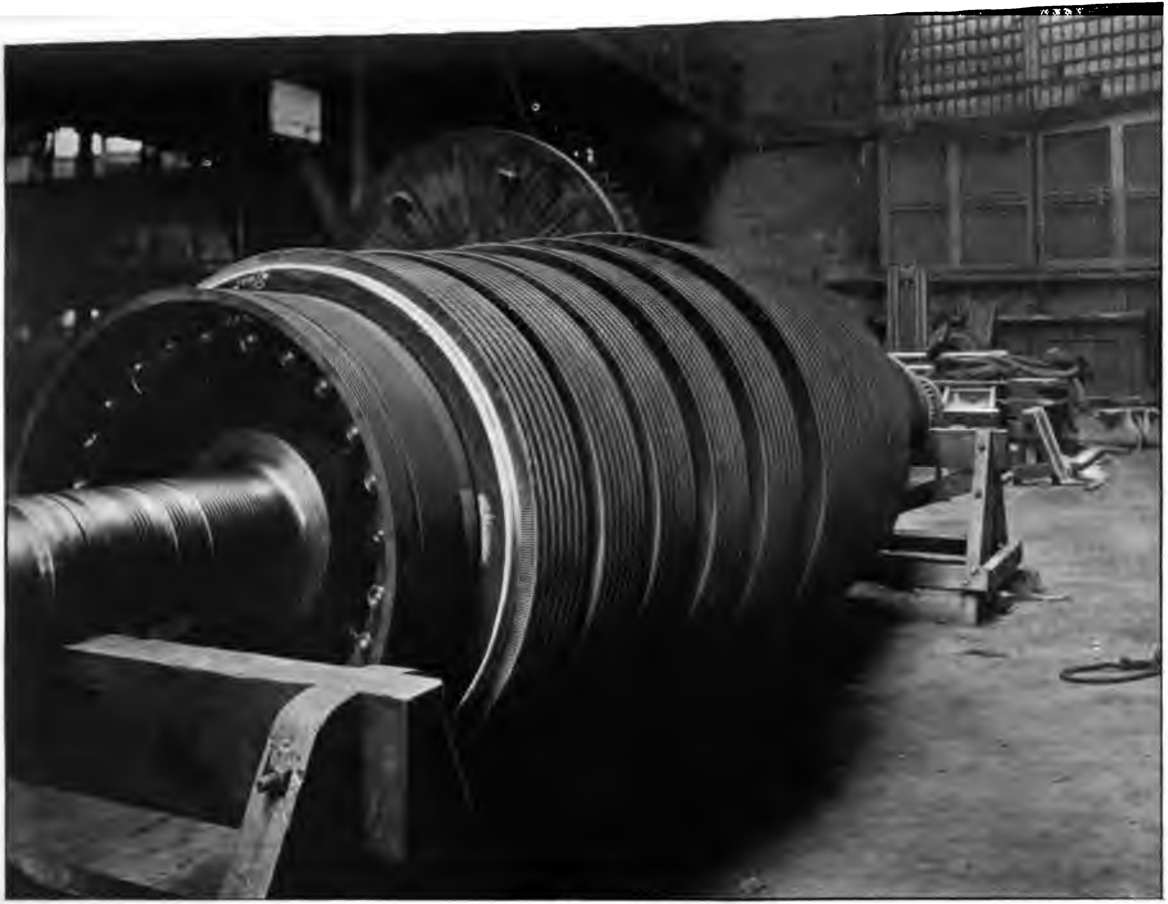
7. Machinery vibration practically eliminated.



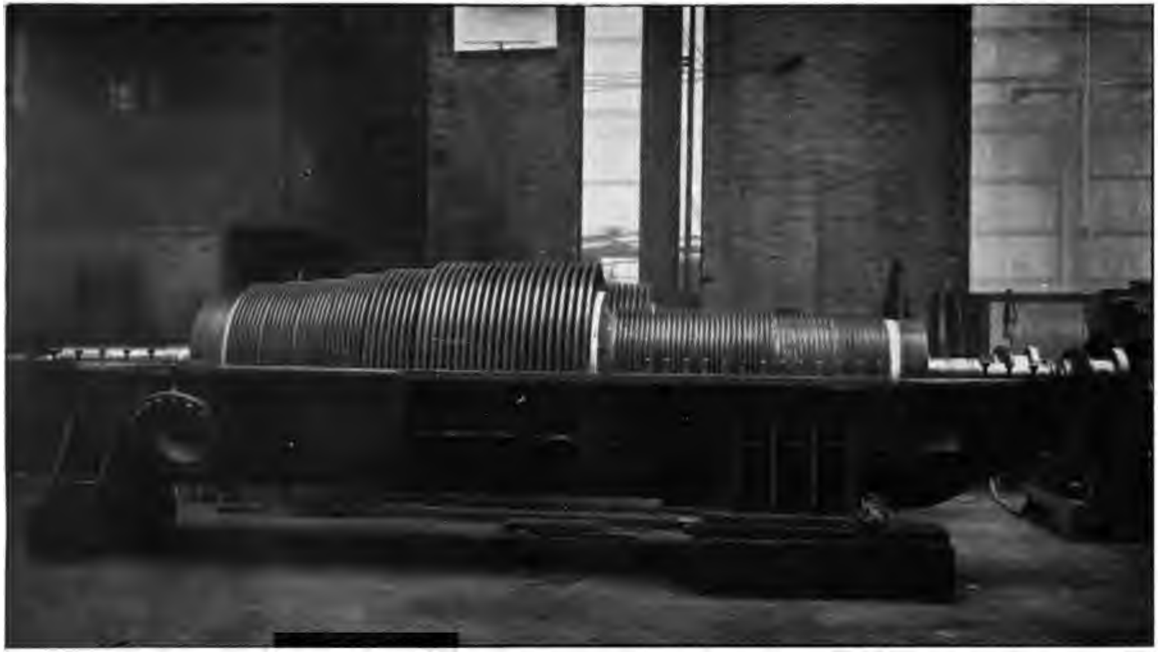
Owners—Messrs G. & J. Burns Ltd. *Builders*—The Fairfield Shipbuilding & Engineering Co. Ltd.

LOW-PRESSURE ROTOR, WITH REVERSE TURBINE COMPLETE, R.M.S. "VIPER."

View showing Thrust Collars, Main Bearings, Gland Collars, Ahead Dummy Piston, Ahead Turbine Expansions, Reverse Turbine Expansions, and Reverse Dummy Piston.

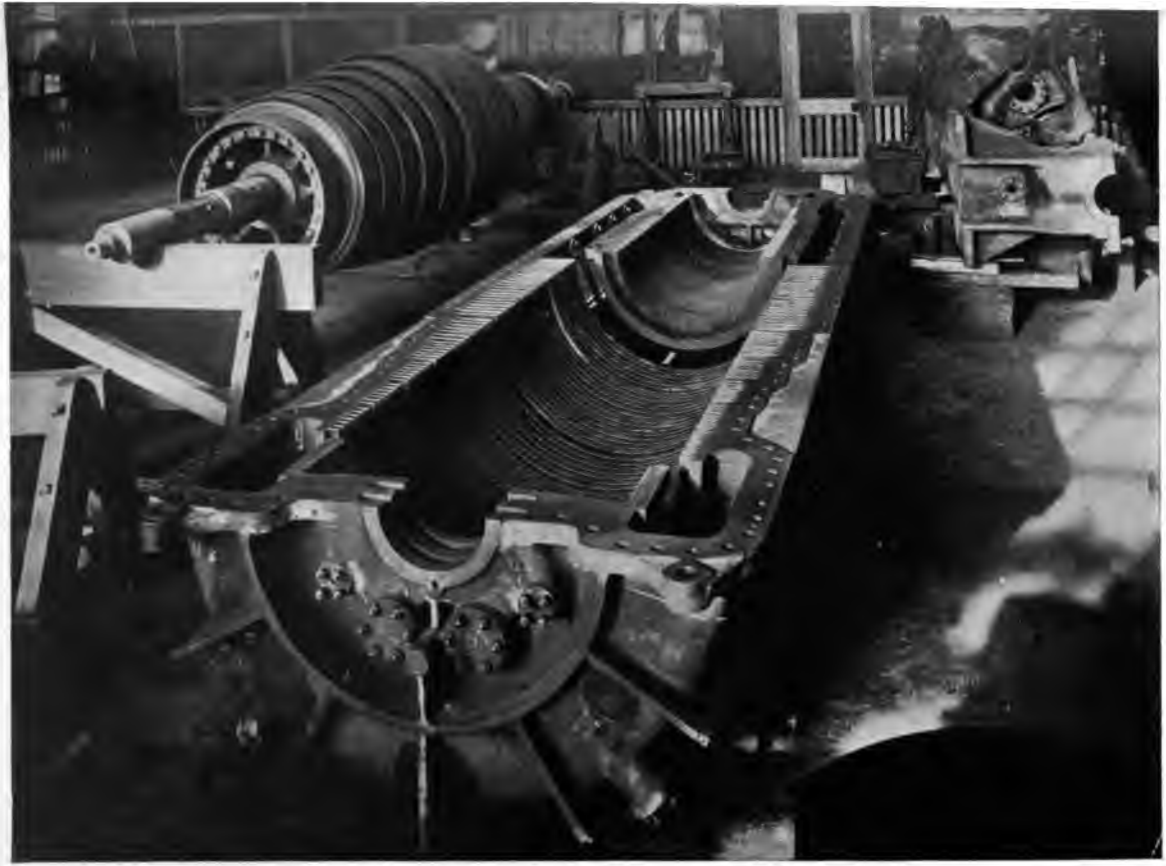


LOW-PRESSURE ROTOR COMPLETE, R.M.S. "VIPER."
View showing Gland Collars, Ahead Dummy Piston, and Steam Expansions.



LOW-PRESSURE ROTOR, IN LOWER HALF CASING, R.M.S. "VIPER."

View showing Ahead Steam Admission, Ahead Dummy Piston, Complete Ahead and Reverse Expansions, also Reverse Dummy Piston.



LOW-PRESSURE TURBINE CASING (UPPER HALF), R.M.S. "VIPER."
View showing Gland Steam Inlet and Outlet, Ahead Dummy Casing, Ahead and Reverse Expansions.

It may, however, be stated that the turbine is most economical in running at high speeds, as the difference in consumption between say 20 knots and 10 knots is not proportional to the difference in speed.

This is due to the fact that the most economical speed of the turbine bears a certain ratio to the velocity of the steam, and if the turbine speed drops below this, the economy falls away.

With regard to the floor space occupied, there is not much to choose between the reciprocating engine and the turbine. In space taken up in the vertical direction, the turbine has a distinct advantage.

Regarding reliability, there is no reason why the turbine should be inferior to the reciprocating engine; and as to economy and speed the turbine has shown itself in many ways superior, although at very low speeds the reciprocating engine is superior in economy of steam; against this, however, the turbine requires less oil. The turbine, in its present state of development, is not suited for all classes of ships. From the weight point of view the turbine is superior to the reciprocating engine of the same power, particularly in the case of high-speed vessels, and possesses also the special advantage of being well balanced.

Although it is undoubtedly the case that many practical improvements have still to be made in the turbine engine, and more particularly in turbine propellers, allowing for its very successful inauguration, and the gigantic advance made in a few years, the writer is of the opinion that the turbine is without doubt the engine of the future. It may be some time yet before sufficient modification of the turbine allows of the adoption of this type of engine for the slow speed tramp steamer, but it is very probable that such modification will come, in perhaps the near future.

It is evident that errors, both of design and construction, have been made in the turbines of large ocean-going steamers, but it is confidently expected that these will be profited by and effectively remedied in the case of the "Express" Cunarders now building.

It is maintained by some that the steam turbine will before long find a keen rival in the gas engine, which is

also being developed to some degree for marine purposes. This may be true to a certain degree, but the gas engine, as applied to marine engineering, has still a very long way to go in development before it can affect in any way the increasingly important position held by the steam turbine at the present time.

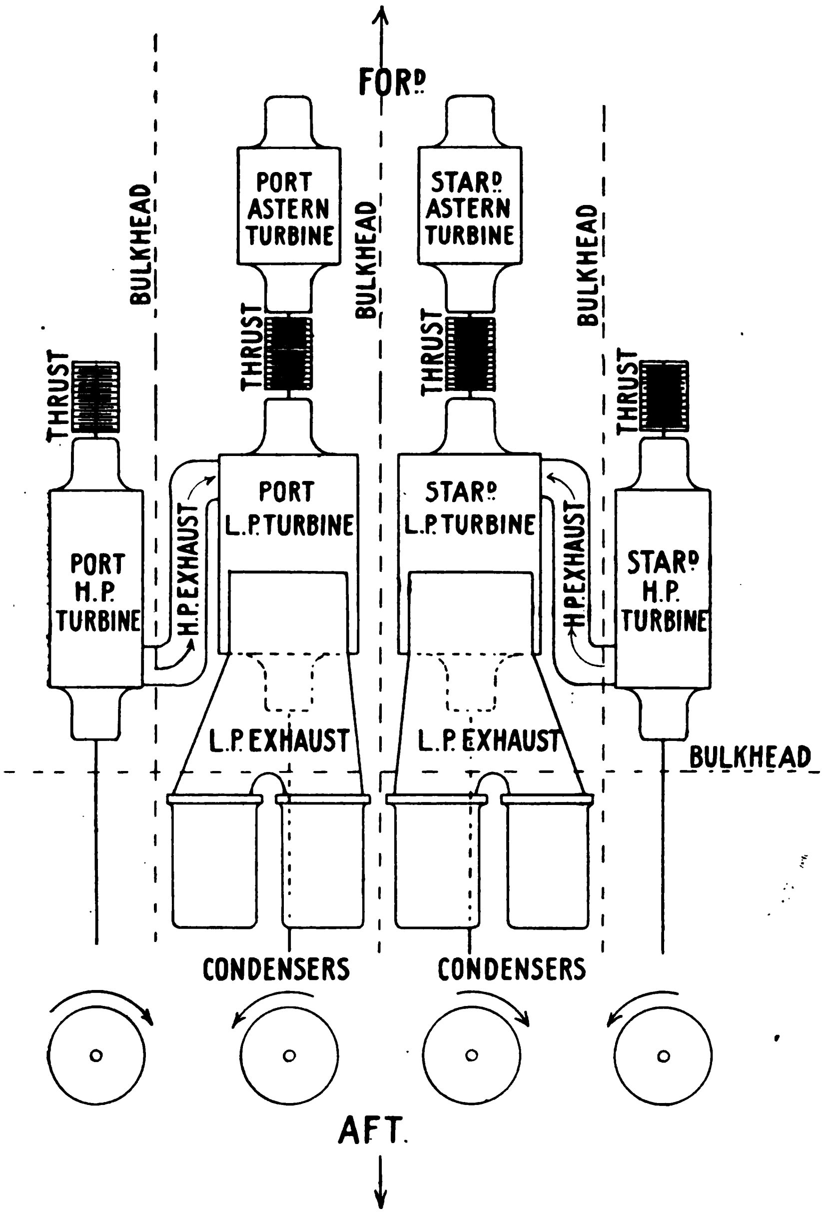
"Carmania."—The following are a few particulars of this, the largest turbine steamer in service, and built by Messrs John Brown & Co., Clydebank, to the order of the Cunard Line:—

Length - - - -	678 feet.
Beam - - - -	72 feet.
Boiler Pressure - - -	195 lbs. per square inch.
No. of Boilers - - -	13.
Revolutions - - -	185 per minute.
Diameter of Propeller . -	14 feet.
Pitch of Propeller - -	13 feet.
Equivalent Horse-Power -	21,000 (at least).
Speed - - - -	21 knots (trial).
Diameter of L.P. Rotor -	138 inches.

On the trial runs made on the Clyde the speed attained by the "Carmania" was 21.6 knots.

NOTE.—The pitch ratio of the propeller is equal to .9, and the area ratio probably about .6 or thereabout.

"Express" Cunarders.—The large Cunarders now under construction for the Atlantic service are fitted with four lines of shafting and four propellers, one to each length. The turbines are six in number, an H.P. on each outside shaft, and an L.P. and reverse turbine on each inside shaft. The reverse turbines are independent of the L.P. turbines, and are placed forward with the thrust block between the two (see sketch). The L.P. rotors are 188 inches in diameter without the blades, the largest height of blade used being 22 inches. The line shafting is 22 inches in diameter, and the rotor spindles 33 inches diameter in the bearings, which are about 5 feet in length. The boilers (cylindrical) are twenty-five in number, working at a pressure of 200 lbs. per square inch. The speed is expected to be 25 knots.



Turbine Arrangement of "Express" Cunarders.

The circles and arrows show the direction of shaft and propeller rotation, which is :—

Port H.P. and Starboard L.P. Right-hand Propellers.
Starboard H.P. and Port L.P. Left-hand Propellers.

Causes of Breakdown.—The principal causes of breakdown in turbine machinery may be classed as follows:—

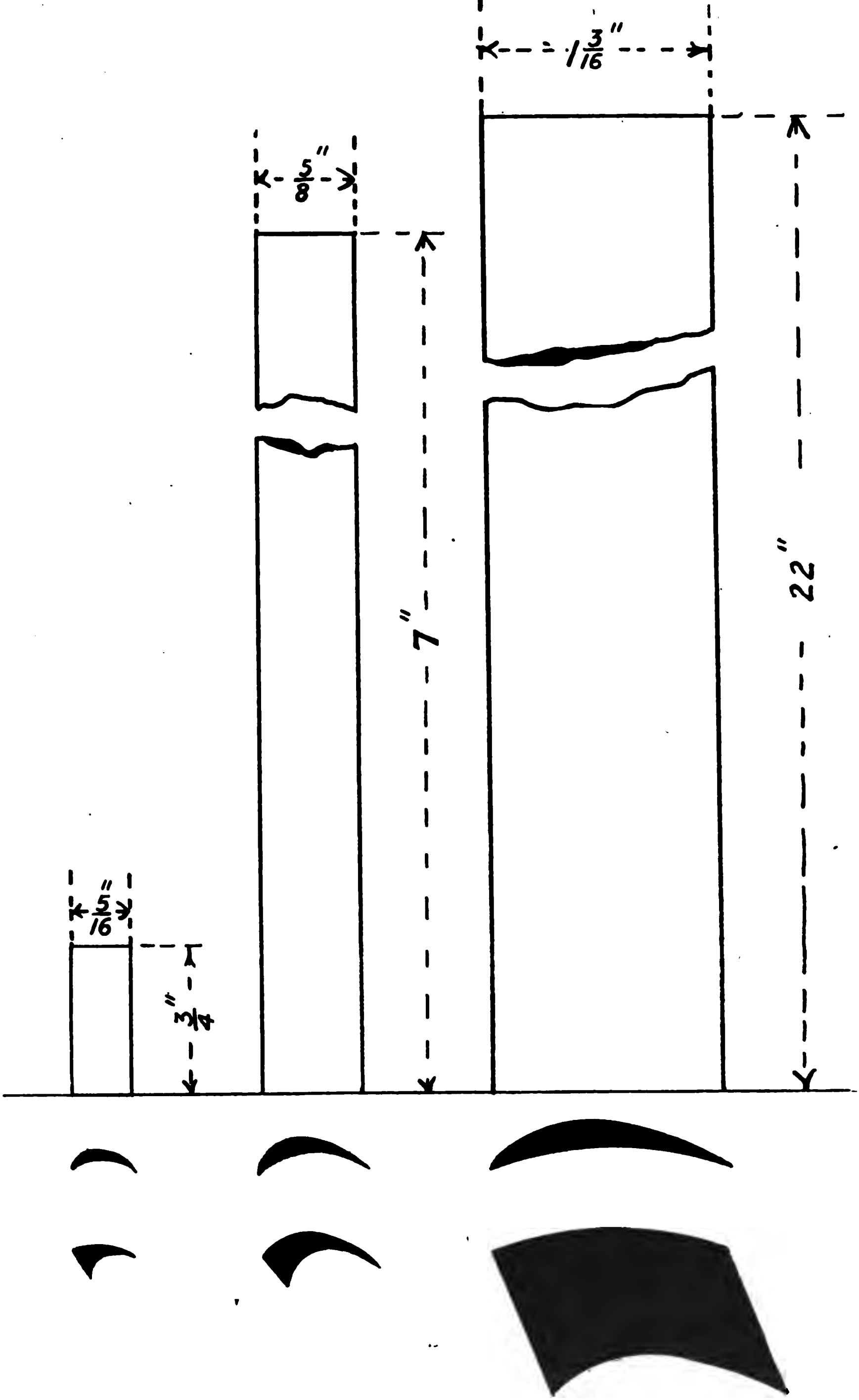
1. **Stripping of Dummy Rings.** — This may be brought about by the (*a*) spindle bearings wearing down; or (*b*) the rotor out of position longitudinally; (*c*) insufficient clearance when heated up.

2. **Stripping of Blades.**—This may be brought about by—(*a*) Wear down of spindle bearings. (*b*) Insufficient clearance between rotor blades and case, or between casing blades and rotor when heated up. It should be noted that the expansion of the rotor blades and rotor when heated up is much more than that of the rotor casing, thus reducing the blade clearance considerably, and increasing the risk of the rotor blades fouling the inside of the turbine casing. (*c*) *Water hammer.*—If a body of water is introduced into the turbine casing through priming in the steam pipes, the blades of the first and second expansion may be shattered by the resulting "water-hammer" action.

Up to the present there have been no cases recorded of "whipping" or "sagging" of the turbine rotor which has resulted in stripped blades, although this breakdown has been confidently predicted by many engineering experts.

The absolute importance of the regular oil supply to each of the rotor spindle main bearings cannot be overestimated, as most of the cases which have actually occurred of dummy ring and blade stripping have been brought about by the temporary stoppage of the oil service, resulting, as may naturally be expected, in rapid heating up of the main bearings, melting of the white metal, and consequent wear down of the rotor, producing as a direct result the fouling of blades and stripping of some of the expansions. It is now the usual practice to insert a sight feed glass in the oil supply pipes, to enable the engineer on watch to see that the oil service is in working order, and the supply pipes all clear.

Corrosion in the Rotor Drums.—One of the most serious troubles yet experienced in marine turbines is that of corrosion inside of the rotor drums. This is undoubtedly due to moisture and oxygen from air introduced



Types of Blades and Packing Pieces (Full Size).

The first is from the 1st H.P. Expansion of a large Channel Steamer, the second is from the last L.P. Expansion of a large Ocean-going Steamer, and the third is from the last L.P. Expansion of the "Express" Cunard Steamers "Lusitania" and "Mauritania."

into the turbine either from the water itself or more likely by leakage at the steam glands, which in the case of the L.P. turbines aft draw in the air, if not blowing out steam.

It should be observed that aft the L.P. turbines work in a high degree of vacuum; the after steam glands thus may draw in air, which, combining with the cast iron, produces by chemical action iron oxide on the inside of the cases. Again, the H.P. turbine runs in a vacuum when the two outside turbines only are running, so that under this condition air may also be drawn into the turbine casing, and corrosion take place.

The insides of the casings are sometimes painted over with composition to check the corrosion referred to.

To prevent the admission of air it is advisable to see that the steam gland outer pocket cock is so regulated as to allow a constant puff outwards of weak steam when the turbines are running.

Wear of Dummy Rings.—In one or two cases it has been found that the dummy rings have been worn away to some considerable extent by the grinding action of chemical matter (iron oxide) produced by corrosion in the dummy cases. This may be assumed to have been brought about by galvanic action set up between the brass rings and cast-iron case when in contact in water, the chemical matter thus formed probably grinding away the rings. The water which accumulates is produced by condensation in the dummy grooves, and constitutes the so-called "water seal" of the dummy. The corrosion referred to is a matter of much importance, affecting as it does the mechanical balance of the turbine, not to mention the resulting loss of economy.

Heating Up.—Previous to starting up a turbine the following precautions should be taken:—

1. Open all turbine casing drains, with "wet" air pumps working slow.

2. Admit steam (reduced) to outer pockets of steam glands until the gauge shows a pound or two pressure.

3. Test dummy clearances before heating up and after heating up. (This is important.)

SECTION III.

DATA FROM PRACTICE, PROPELLERS, CONSUMPTION, &c.

Blade Dimensions.—The following dimensions, taken from the L.P. turbines of a large ocean-going steamer, will afford a fair idea of the proportions between the blade widths, blade heights, and blade tip clearances adopted in actual practice. The L.P. turbines consist of 8 expansions arranged as follows :—

L.P. Turbines.

Rotor Drum, 7 ft. 9 in. Diameter.

Expansion.	Blades.	Blade Tip Clearances (Cold).
1st	10 Rows of $\frac{3}{8}''$ blades $1\frac{3}{8}''$ high and $1\frac{1}{8}''$ pitch	.08″
2nd	10 „ $\frac{3}{8}''$ „ $2''$ „ $1\frac{1}{4}''$ „	.08″
3rd	10 „ $\frac{3}{8}''$ „ $2\frac{3}{4}''$ „ $1\frac{3}{8}''$ „	.08″
4th	10 „ $\frac{3}{8}''$ „ $4''$ „ $1\frac{1}{2}''$ „	.09″
5th	10 „ $\frac{1}{2}''$ „ $5\frac{1}{2}''$ „ $1\frac{7}{8}''$ „	.10″
6th	10 „ $\frac{1}{2}''$ „ $7''$ „ $2''$ „	.12″
7th	10 „ $\frac{1}{2}''$ „ $7''$ „ $2''$ „	.12″
8th	10 „ $\frac{1}{2}''$ „ $7''$ „ $2''$ „	.12″

Dummy clearance, .03″ (or $\frac{30}{1000}''$).

Observe that the blades of the 6th, 7th, and 8th expansions are all of the same height and pitch, the only difference

being that of blade section, the 7th and 8th expansions having blades of a flatter surface section and a greater circumferential pitch, as the packing pieces are thicker.

In steamers of from 5,000 to 8,000 estimated I.H.P. the blades vary in width from about $\frac{5}{16}$ in. in the first H.P. expansion to $\frac{1}{2}$ in. in the L.P. expansions, and in height from $1\frac{1}{16}$ in. in the H.P. turbine to about 8 in. or 10 in. in the L.P. turbines, depending on the diameter of rotor drum and the number of revolutions. In general, the larger the rotor drum diameter the shorter the blade heights.

Number of Expansions.—The usual number arranged is 4 expansions in the H.P. turbine, and 8 expansions in each L.P. turbine, the total number of rows of blades being the same for each separate turbine. So that if the H.P. turbine is made up of say 4 expansions, each containing 16 rows of blades, then each L.P. turbine will have 8 expansions, each containing 8 rows of blades. Or again, suppose the H.P. turbine to consist of 4 expansions with 14 rows of blades in each, then the L.P. turbines will consist of 8 expansions with 7 rows of blades in each. It must of course be understood that the rotors will in this case contain 56 rows in all, and the casings 56 rows in all; or 56 rows of moving blades and 56 rows of fixed blades for each turbine, H.P. and L.P. A pair of rows, consisting of one row of fixed and one row of moving blades, is usually called a "stage." So that in the above case the H.P. turbine will have 14 stages in each expansion, and the L.P. turbines have 7 stages in each expansion.

The "packing pieces," which are caulked in between the blades, vary in shape according to position, being of small thickness section and much curved at the first expansions, and of larger thickness section and less curvature at the last expansions. The shape of the packing piece exactly corresponds to the two surfaces of the blade which fits in between. So that there is a much smaller curve on the side in contact with the steam-receiving surface of the blade than that in contact with the convex surface or back of the blade.

Number of Blades per Row.—At each successive expansion, as the size of the packing pieces gradually increases, there will necessarily be a smaller number of blades in one complete row or ring. So that if the first expansion contains say 100 blades per row, the last expansion may only contain about half that number, owing to the increased thickness of the packing pieces required to allow (together with the increased blade heights) for the increase of steam volume at the exhaust end of the turbine.

The number of blades per row of rotor and casing thus varies throughout the turbine in a diminishing ratio from the initial to the exhaust end.

Blade Expansion by Heat.—As the largest blades are situated at the lowest pressure and temperature end of the turbine, they are exposed to less heat than the smaller blades at the admission end, and consequently less expansion clearance may be allowed without danger of the blade tips making contact with the inside of the casing or the outside of the rotor drum.

In heating up to the working conditions of actual service it is found that the average expansion (due to heat) is about $\frac{1}{1000}$ in. per inch height of blade at admission end of turbine, and less than this at the exhaust or low-pressure end.

The minimum or ahead working clearance between the dummy rings and grooves is usually set for about $\frac{25}{1000}$ to $\frac{30}{1000}$ of an inch, and the maximum or astern working clearance about $\frac{10}{1000}$ more, or say $\frac{40}{1000}$ of an inch. This is due to the slight change in rotor position aft which takes place in bringing up the astern or reverse surfaces of the thrust rings and collars in contact when the direction of rotation is reversed.

Record of Wear Down at Main Turbine Bearings.

NOTE.—The following figures refer to the wear down as measured by the "bridge gauge" (see page 60). The

first column shows the actual gauge clearance when new, and the other column the subsequent increase of clearance due to wear down when tested one year later.

	Original Marking.	After One Year.
H.P. Forward - -	.033	.034 (tight)
H.P. Aft - - -	.036	.035
Port L.P. Forward - -	.041	.044 (tight)
Port L.P. Aft - - -	.04	.045
Starboard L.P. Forward -	.037	.044
Starboard L.P. Aft - -	.032	.036

Tip Clearance.—To allow of rotor and blade expansion under heat a certain working clearance has to be allowed between the tips of the rotor blades and the inside surface of the casing, also between the tips of the casing blades and the outer surface of rotor drum. This of course works out as a certain per cent. of loss by leakage over blade tips, no useful work being done by the steam in passing through.

It is calculated that if the loss is 3 per cent. when the revolutions are 600 per minute, then the loss at 200 revolutions per minute will be 27 per cent., as the per cent. loss varies as the square of the revolutions, or, which is the same thing, as the square of the rotor diameter.

So that, allowing for the increased expansion due to larger drums, the loss increases by the square of the diameter, or in other words, by the area open to leakage.

Clearances.—The following typical examples of blade tip clearances and dummy clearances were taken from the turbines of a large channel steamer:—

BLADE TIP CLEARANCES.

Taken when cold.

H.P. Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.041	.041	.052	.043	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{15}{64}$	$\frac{7}{32}$
„ 2	.051	.055	.059	.049	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
„ 3	.063	.055	.070	.061	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{1}{4}$
„ 4	.049	.045	.054	.051	$\frac{23}{64}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$

Starboard L.P. Turbine.

Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.070	.080	.068	.070	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{13}{64}$	$\frac{17}{64}$
„ 2	.072	.085	.070	.085	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$
„ 3	.078	.090	.082	.092	$\frac{11}{32}$	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{17}{64}$
„ 4	.082	.092	.082	.085	$\frac{3}{8}$	$\frac{11}{32}$	$\frac{5}{16}$	$\frac{3}{8}$
„ 5	.085	.093	.085	.088	$\frac{7}{16}$	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{11}{32}$
„ 6	.095	.123	.098	.115	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{7}{16}$
„ 7	.102	.125	.105	.112	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{15}{32}$
„ 8	.102	.115	.105	.113	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{2}$

Starboard Astern Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.055	.062	.068	.070	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 2	.065	.064	.080	.082	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{7}{32}$	$\frac{17}{32}$
„ 3	.098	.102	.122	.128	$\frac{3}{16}$	$\frac{17}{32}$	$\frac{3}{16}$	$\frac{17}{32}$
„ 4	.098	.100	.125	.110	$\frac{7}{32}$	$\frac{17}{32}$	$\frac{7}{32}$	$\frac{17}{32}$

	Cold Setting.	On Trial.
H.P. Dummy Clearance - -	$\frac{15}{1000}$	$\frac{30}{1000}$
S.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{21}{1000}$
P.L.P. „ „ - -	$\frac{36}{1000}$	$\frac{27}{1000}$

It will be noted that the H.P. Dummy clearance *decreased* when heated up, and the L.P.'s increased when heated up.

Port L.P. Turbine.

Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.069	.070	.075	.075	$\frac{15}{64}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{32}$
„ 2	.072	.070	.088	.085	$\frac{17}{64}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{4}$
„ 3	.070	.075	.078	.088	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{16}$
„ 4	.070	.078	.083	.092	$\frac{9}{32}$	$\frac{17}{64}$	$\frac{3}{8}$	$\frac{5}{16}$
„ 5	.072	.073	.091	.090	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{5}{16}$
„ 6	.102	.099	.122	.123	$\frac{13}{32}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$
„ 7	.108	.108	.128	.115	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{27}{64}$	$\frac{7}{16}$
„ 8	.117	.118	.132	.148	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{31}{64}$	$\frac{15}{32}$

Port Astern Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	.085	.085	.067	.065	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{2}$
„ 2	.085	.088	.073	.073	$\frac{1}{8}$	$\frac{17}{32}$	$\frac{11}{64}$	$\frac{17}{32}$
„ 3	.115	.113	.112	.098	$\frac{1}{8}$	$\frac{1}{2}$	$\frac{5}{32}$	$\frac{15}{32}$
„ 4	.120	.115	.115	.103	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{7}{32}$	$\frac{1}{2}$

Observe that the reverse turbine longitudinal clearance is much more *aft* than forward, to allow of the heating up and expansion of the drum and casing, which takes effect on the after end of the turbine.

Turbine Pressure Data, &c.

(From actual practice.)

(1.) The following results obtained in trial runs of the Isle of Man passenger steamer, "Manxman," give a good idea of the figures usually obtained in turbine steamers of this class:—

Mean speed of two runs - - - - -	23.14 knots.
Boiler pressure per square inch - -	192 lbs.
Steam in high-pressure turbine - - -	180 „
„ low-pressure turbine, port - -	20 „
„ „ „ starboard -	20 „
Vacuum in condenser, port - - -	28.25 in.
„ „ starboard - -	28.4 „
Revolutions per minute, high-pressure turbine	533
„ „ low-pressure „	609
Temperature of feed-water leaving heater -	180° Fahr.
Air-pressure in stokehold - - - - -	1.5 in.

(2.) The following data refer to a cross-channel steamer:—

Full Speed Ahead (all Turbines).

Speed - - - -	22 knots.
I.H.P. - - - -	6,500.
Boiler - - - -	160 lbs.
H.P. turbine - - -	140 „
L.P. port - - - -	20 „
L.P. starboard - - -	20 „
L.P. port (astern) - -	23 in. vacuum.
L.P. starboard (astern) -	23 „ „
Condenser vacuum - -	$24\frac{1}{2}$ „ „
Revolutions - - -	630 per minute (all shafts).
Propeller pitch - - -	4 ft. 6 in.
Slip - - - - -	21 per cent.

(3.) **Gauge Indications.**—The following figures from actual practice give a fair idea as to the usual pressures and vacuum carried in the turbines of a steamer of moderate power:—

Type—Channel Steamer.

Boiler pressure - - - -	160 lbs.
H.P. turbine (initial) - - -	150 „
L.P. ahead turbines (initial) - -	24 „
Astern turbines of L.P. shafts - -	25 in. vacuum.
Condenser - - - - - -	27 „ „
Revolutions - - - - -	650 (all shafts).

NOTE.—When running ahead at reduced speed with *all turbines* working, the H.P. pressure is 80 lbs., L.P. pressures 15 lbs. pressure, and condenser 28 in. vacuum.

Type—Channel Steamer.

(4.) **Running Full Power Ahead with all Three Turbines.**

Boiler pressure - - - -	155 lbs. (gauge).	
H.P. turbine pressure - - -	146 „ „	
Port L.P. turbine pressure - -	13 „ „	
Starboard L.P. turbine pressure -	$12\frac{1}{2}$ „ „	
Condenser vacuum - - - -	27 in.	
H.P. forward steam gland pressure -	1 lb.	These steam glands have no "leak-off" connections.
H.P. aft steam gland pressure - -	1 „	
L.P. forward steam gland pressures -	2 lbs.	
L.P. aft steam gland pressures - -	$3\frac{1}{2}$ „	
Coal per shaft horse-power per hour -	1.8 „	

NOTE.—Observe that the L.P. turbine steam gland pressures are *higher* than the H.P. turbine gland pressures, this being required to prevent the admission of air into the turbine casing, which would destroy the L.P. vacuum.

(5.) Running Ahead with Two Outside (L.P.) Turbines only.

(*Centre shaft running idly in vacuum at about $\frac{1}{3}$ revolutions of outside shafts.*)

Speed - - - - - - - -	15 knots.
H.P. turbine pressure - - - -	20 in. vacuum.
L.P. turbine pressures, port and starboard -	20 lbs. (gauge).
Condenser vacuum - - - - - -	$27\frac{1}{2}$ in.

(6.) Running Full Speed Astern with Two Outside (L.P.) Turbines.

Speed - - - - - - - -	14 knots.
H.P. turbine pressure - - - -	20 in. vacuum.
L.P. turbine pressures, port and starboard -	80 lbs. (gauge).
Condenser vacuum - - - - - -	27 in.
Revolutions - - - - - -	500.

NOTE.—When running ahead with all turbines one inch increase of vacuum gave 23 extra revolutions to each L.P. turbine shaft. The importance of a high degree of vacuum will thus be obvious.

In the steamer referred to above, the pitch of the outside propellers is 4 ft. 3 in., and of the H.P. (centre) 4 ft. 6 in. This difference of pitch is accounted for by the fact that the centre or H.P. shaft runs at less revolutions than the outside shafts, and therefore necessitates a larger pitch to obtain the same screw and ship advance. An example will make this clear.

Speed - - - - - - - -	20 knots.
Slip - - - - - - - - -	28 per cent.
L.P. turbine revolutions - - - -	662 per min.
H.P. turbine revolutions - - - -	620 ,,

To find the required pitch of propellers:—

Knots × 6080 = Pitch × Revolutions × 60 × Effective % advance.

Therefore, $20 \times 6080 = \text{Pitch} \times 662 \times 60 \times .72$,

and, $\dfrac{20 \times 6080}{662 \times 60 \times .72} = 4.2$ feet pitch (outside propellers),

and, $\dfrac{20 \times 6080}{620 \times 60 \times .72} = 4.5$ feet pitch (centre propeller).

NOTE.—$100 - 28 = 72$, and $\frac{72}{100} = .72$ efficiency of screw.

Consumption.—The consumption for this steamer worked out as 1.8 lb. of coal per *shaft horse-power* per hour, the power being calculated by means of the Denny & Johnson torsion meter (see page 141). This consumption, if reduced to the corresponding indicated horse-power, would be less per horse-power in proportion, as the shaft or transmitted horse-power is obviously less than what the indicated horse-power would be, were it possible to indicate the power.

The slip at full ahead speed worked out as 30 per cent. which, at an ahead speed of from 10 to 14 knots, fell off to 11 per cent.

"Steam Gland" Connections.—Sometimes this cock, fitted to the end of the turbine casing at the "steam gland" port, or "outer pocket," is of the two-way type, and is arranged either to (1) give direct steam to the gland, or (2) connect the gland with the condenser. This is necessary in the case of the H.P. steam glands, as, when the two outside shafts are working only, the H.P. turbine is then merely revolving inertly in a vacuum, due to the water action on the screw. Under these conditions the cocks can be opened to the condenser, but when the H.P. is running ahead with full steam, the cocks can then be opened to admit a low steam pressure to the glands (see sketch).

NOTE.—With the above arrangement the "leak-off" port and pipe connection (see page 78) are both omitted.

Vacuum in Idle Turbines.— When running ahead with all three turbines the reverse turbines are revolving in a vacuum of from 22 to 25 in., and when running ahead or reverse with the outside shafts only, the centre or H.P. turbine is then revolving in a vacuum also of about 25 in. In this case the centre propeller is caused to revolve by the action of the water on the blades, being forced round in the ahead direction if the outside shafts are running ahead, and forced round in the astern direction if the outside shafts are running astern at the time.

Importance of High Vacuum.—A high intensity of vacuum is of considerable importance to a turbine, as the

total expansions of steam may be taken to vary in proportion to the amount of vacuum obtained.

Suppose $\begin{cases} \text{Initial pressure in H.P. turbine} = 140 \text{ lbs. gauge,} \\ \text{Vacuum in condenser} = 26 \text{ in.,} \end{cases}$

the actual vacuum at the last L.P. turbine row will probably be somewhat less, say 24 in., so that

$$24 \text{ in.} \div 2 = 12 \text{ lbs.,}$$

$$\text{and } 15 - 12 = 3 \text{ lbs. gross terminal pressure.}$$

$$\text{Therefore, } (140 + 15) \div 3 = 51.6 \text{ expansions of steam in all.}$$

If therefore the vacuum is increased to say 28 in. in condenser and 26 in. at L.P. turbine exhaust the result will be as follows:—

$$26 \text{ in.} \div 2 = 13 \text{ lbs.,}$$

$$\text{and } 15 - 13 = 2 \text{ lbs. gross terminal pressure.}$$

$$\text{Therefore, } (140 + 15) \div 2 = 77.5 \text{ expansions of steam in all.}$$

From the foregoing it will be evident that the economy on turbine engines depends greatly on the vacuum carried.

(7.) Type—Channel Steamer.

(9,000 Horse-Power.)

Speed Trials (Ahead).

Revolutions per Minute (mean of three shafts).	H.P. Turbine Pressure.	Port L.P. Ahead Turbine Pressure.	Starboard L.P. Ahead Turbine Pressure.	Speed in Knots.
290	10 lbs.	21 in. vacuum	21 in. vacuum	10.68
409	35 „	12 „ „	12 „ „	14.73
512	60 „	0	0	18.12
608	115 „	22 lbs.	22 lbs.	20.82
680	152 „	23 „	23 „	22

(8.) Astern Trials (Two L.P. Turbines only).

H.P. Turbine Pressure.	Port L.P. Turbine Pressure.	Starboard L.P. Turbine Pressure.
18 in. vacuum	82 lbs.	80 lbs.
25 " "	50 "	50 "

NOTE.—Average revolutions 540 per minute, and mean astern speed 14.4 knots, at higher L.P. pressure shown above.

(9.) **Turbines in Vacuum.**—(1) When running ahead with all three turbines the reverse (L.P.) turbines are revolving inertly in a vacuum.

(2) When running ahead with the two outside turbines only, the H.P. turbine and the two reverse (L.P.) turbines are revolving in a vacuum.

(3) When running astern with the two outside turbines, the two L.P. ahead turbines and the H.P. turbine are revolving in a vacuum.

The average vacuum indications in a case noted by the writer were as follows:—

In No. (1), reverse turbine vacuum, 24 to 26 in.
" (2), H.P. vacuum, 25 to 27 in.
" (3), L.P. ahead turbines, 26 to 27½ in.

(10.) **Steam Gland Pressures.**—With the steam gland and leak-off arrangements as presently fitted at each end of the turbines the pressures indicated, when running ahead full power, were as follows:—

	Outer Pocket (Steam Inlet).	Inner Pocket (Leak-off to L.P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	1 lb.	10 to 15 in. vacuum
L.P. Turbine (forward and aft glands) -	1 "	5 to 15 " "

The "steam to glands" pipe is led off the main steam pipe, and is reduced in pressure by throttling the cock or valve to about 40 lbs.; this is, of course, still further reduced at the "outer pocket" of the glands by wire-drawing at the cock on the gland itself.

The inner pocket of the steam glands leaks off the small quantity of steam which finds its way past the brass rings and collars to the 3rd expansion of the L.P. turbines. All the leak-off connections from all three turbines forward and aft are connected up in this way when running ahead with all three turbines. When running ahead or astern with the two outer (L.P.) turbines only, the H.P. leak-off cocks are changed over direct to the condenser with the pressure results shown below.

(11.) **Running Ahead with Two L.P. Turbines only.**

	Outer Pocket (open to Condenser).	Inner Pocket (Leak-off to L.P. 3rd Expansion).
H.P. Turbine (forward and aft glands) -	27 in. vacuum	5 lbs. steam pressure

NOTE.—If the speed is reduced when running ahead with all three turbines, the "outer pocket" steam inlet pressure increases by a few pounds, and the "inner pocket" leak-off vacuum also increases by a few inches.

It should also be noted that the H.P. steam gland cocks are of the "two-way" type, to allow of the change over to condenser when required; whereas the L.P. steam gland cocks are only "one-way" type. The "leak-off" cocks from all turbines are single-way cocks.

(12.) **Turbine Dimensions.**—The following are the principal turbine dimensions of a large cross-channel steamer of about 10,000 I.H.P., speed 22 knots, and

revolutions about 500, the propeller pitch being 5 ft. 6 in.:—

	H.P. Rotor.	L.P. Rotor.	Reverse Rotor.
Diameter - - - - - - -	48 in.	68 in.	48 in.
Length - - - - - - -	68 „	90 „	58 „
Number of Expansions - - -	4	8	4
Number of Blade Rows in each Expansion	12	6	12
Total number of Rows of Blades -	48	48	48
Blade heights in each Expansion—			
1st Expansion - - - -	$1\frac{3}{8}$ in.	$1\frac{3}{8}$ in.	$\frac{3}{4}$ in.
2nd „ - - - - -	2 „	2 „	$1\frac{1}{2}$ „
3rd „ - - - - -	$2\frac{3}{4}$ „	$2\frac{3}{4}$ „	3 „
4th „ - - - - -	4 „	4 „	3 „
5th „ - - - - -	...	$5\frac{1}{2}$ „	...
6th „ - - - - -	...	8 „	...
7th „ - - - - -	...	8 „	...
8th „ - - - - -	...	8 „	...

NOTE.—It should be noted that the casings will contain the same number of blade rows as the respective rotors, that is, 48 rows in each.

Blade Tip Leakage Area.—The leakage over blade tips depends on the clearance between the rotor blades and the inside of the case, and the clearance between the casing blades and the outside of the rotor. The so-called "blade height" includes the two working clearances referred to, and when these are deducted the effective heights of the blades are correspondingly reduced. To take an example:—

Rotor diameter, outside, 48 in.
Blade height at 1st expansion, $1\frac{1}{16}$ in.
Tip clearance, $\frac{50}{1000}$ in., or .05 in.

From the foregoing it will be obvious that the effective blade height is equal to $1\frac{1}{16}$ in. $-.05 \times 2$ in. $= .9625$ in., and the *mean* diameter of blades 48 in. $+ 1\frac{1}{16}$ in. $= 49\frac{1}{16}$ in. To find the per cent. leakage over blade tips, if the blade

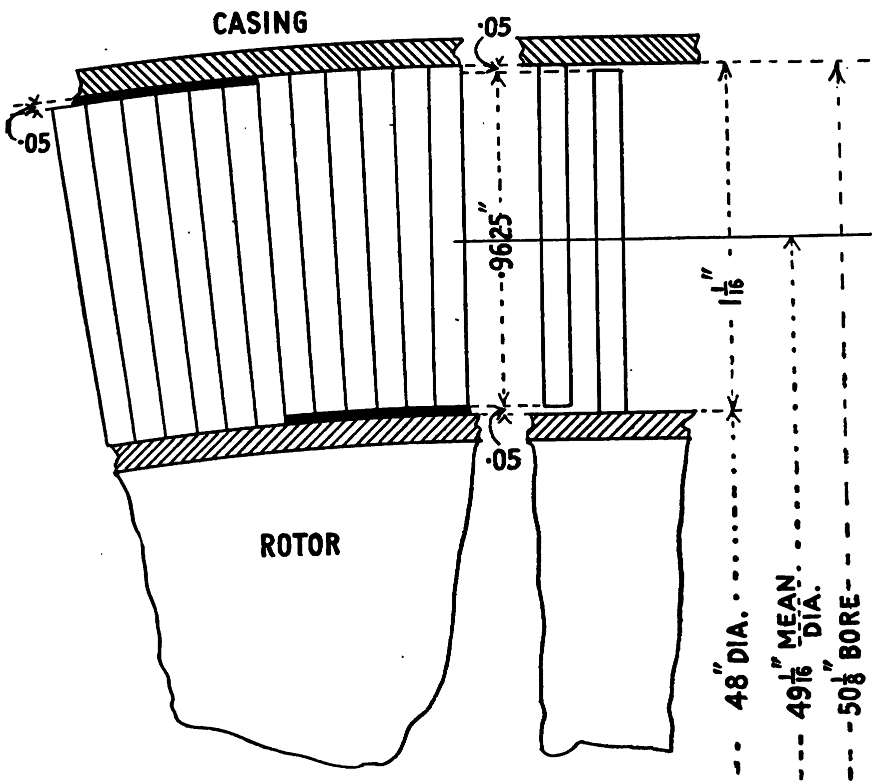
openings at exit edge are equal to .3 of annular area formed by blades, then—

$$49.0625 \times 3.1416 \times .9825 \times .3 = 45.4 \text{ sq. in. clear area,}$$

$$\text{and } 49.0625 \times 3.1416 \times .05 \times 2 = 15.413 \text{ sq. in. leakage area.}$$

$$\text{Therefore, } \frac{15{\cdot}413 \times 100}{45{\cdot}4} = 33{\cdot}9 \text{ per cent.}$$

NOTE.—$49\frac{1}{16}$ in. = 49.0625 in.



Tip Clearance Areas.

NOTE.—The Black Section represents area open for leakage.

The tip clearance area is therefore equal to about 34 per cent. of the steam area through blades.

It does not appear to be definitely known yet whether the steam which leaks over the blade tips is really all lost or not, some contending that a superheating effect is pro-

duced on the steam in the succeeding expansions. The necessity for reducing the blade tip clearances to the smallest amount possible with working requirements will be easily seen from the foregoing example.

NOTE.—The foregoing calculations only refer to the tip clearances *when cold;* when the turbine is heated up under working conditions the clearances are much less.

Propellers.

Number of Propellers.—Three propellers are usually fitted, one on each shaft, as this arrangement has been found to give the most satisfactory results as regards vibration aft just over the propellers.

NOTE.—In the large "Express" Cunard liners now under construction, four lines of shafting and four propellers are being employed. (See sketch, page 95.)

Turbine Propeller Balance.—Turbine propellers are carefully balanced on knife edges, a spindle passing through the hole in the boss for that purpose when testing. The difference in weight is made up or taken away on the astern or forward surface of the blade. The thrusting surfaces of the blades are then carefully chipped and highly polished, as the decided advantage obtained by this finish in reducing skin friction on the blades is now generally recognised.

Cones.—Turbine propellers are always fitted with long tapering cones to allow of the water from the blades having a clear run aft, and thus offering little or no resistance to the thrusting column of water.

Owing to the high revolutions necessary to obtain suitable turbine efficiency, the pitch of the propellers is small. An example will make this clear.

EXAMPLE.—Speed 17 knots, revolutions 400, and slip (assumed) 22 per cent.; find the required pitch of the turbine propellers.

$$\text{Then, } \frac{17 \times 6080}{400 \times 60 \times .78} = 5.5 \text{ feet Pitch.}$$

NOTE.—$100 - 22 = 78$ per cent., and $\frac{78}{100} = .78$ efficiency of screw.

The diameter of the propellers is also necessarily much less than usual, owing to limited space aft, and to obtain the required blade area a much larger ratio of blade surface to disc surface is developed. This fact chiefly explains the somewhat peculiar shape of blade met with in the majority of turbine propellers (see sketches), all of which are much broader in proportion to length than ordinary propeller blades.

It is sometimes found necessary during the period of the steamer's trial runs to alter the propeller blades, giving more or less surface as may be found by results to be beneficial, before the required speed can be obtained. This indicates that calculations relating to propeller design do not always give the results desired in practice, and emphasises the need for still more reliable information as to the true action of the marine screw propeller, and, if at all possible, for some standard method of accurate design, which will obviate the necessity for costly "trial and error" experiments with blades of various surfaces and contour.

Pitch Ratio. — In ordinary marine reciprocating practice the propeller, the "pitch ratio," or pitch divided by the diameter, gives a number varying from 1 to 1.4, but in nearly all turbine practice the pitch ratio works out as a decimal figure, such as .8 or .9; in other words, the pitch is less than the diameter instead of being more.

Area Ratio.—The area of the propeller tip circle divided into the total expanded blade area is known as the "expanded area ratio," and is usually from .3 to .4 in reciprocating engine practice, but in turbine practice the area ratio varies from .4 to .8, owing to the necessity for crowding the required blade surface into the small circle described by the propeller diameter.

EXAMPLE.—The pitch of an ordinary propeller is 15 ft. and the diameter 12 ft., find the "pitch ratio."

$$\text{Then, } 15 \div 12 = 1.25 \text{ Pitch Ratio.}$$

EXAMPLE.—The pitch of a turbine propeller is 8 ft. and the diameter 9 ft., find the "pitch ratio."

$$\text{Then, } 8 \div 9 = .88 \text{ Pitch Ratio.}$$

EXAMPLE.—The diameter of an ordinary propeller is 12 ft. and the expanded blade area 34 sq. ft., find the "expanded area ratio."

$$\text{Then, } 34 \div 12^2 \times .7854 = .3 \text{ Expanded Area Ratio.}$$

EXAMPLE.—The diameter of a turbine propeller is 5 ft. and the expanded blade area 12 sq. ft., find the "expanded area ratio."

$$\text{Then, } 12 \div 5^2 \times .7854 = .6 \text{ Area Ratio.}$$

NOTE.—The *projected* blade area is the actual area of blades as seen from aft looking forward, and constitutes the effective "thrusting area" of the propeller. The "projected area ratio" is required in propeller design calculations.

Slip per cent.—In turbine steamers the apparent slip varies greatly with speed variation and weather conditions. A typical case is given below.

EXAMPLE.—At a speed of 21 knots the revolutions of the turbines of a channel steamer are 635 per minute; to find the slip per cent. if the propeller pitch is 4 ft. 6 in.

$$\text{RULE.—}\frac{\text{Pitch} \times \text{Revolutions} \times 60}{6080} = \text{propeller speed.}$$

$$\text{Therefore, } \frac{4.5 \times 635 \times 60}{6080} = 28.2 \text{ knots.}$$

$$\text{Then, } 28.2 - 21 = 7.2 \text{ knots slip.}$$

$$\text{And, } \frac{7.2 \times 100 \text{ per cent.}}{28.2} = 25.3 \text{ per cent. slip (apparent).}$$

Pitch Variation.—Turbine propeller blades generally present true screw surfaces, as no gain or advantage has been discovered by the adoption of pitch variation, either radially or peripherally, the all important factors in propeller design consisting of the correct adjustment of pitch, diameter, and surface. At the same time, it should be mentioned that the propeller blades of the "Express" Cunarders are constructed with a peripheral pitch variation, the mean pitch being somewhere about 16 ft. This can be shown by the following: Speed 25 knots, revolutions 190, slip (assumed) 15 per cent.

$$\text{Then, Pitch} = \frac{25 \times 6080}{190 \times 60 \times .85} = 16 \text{ feet.}$$

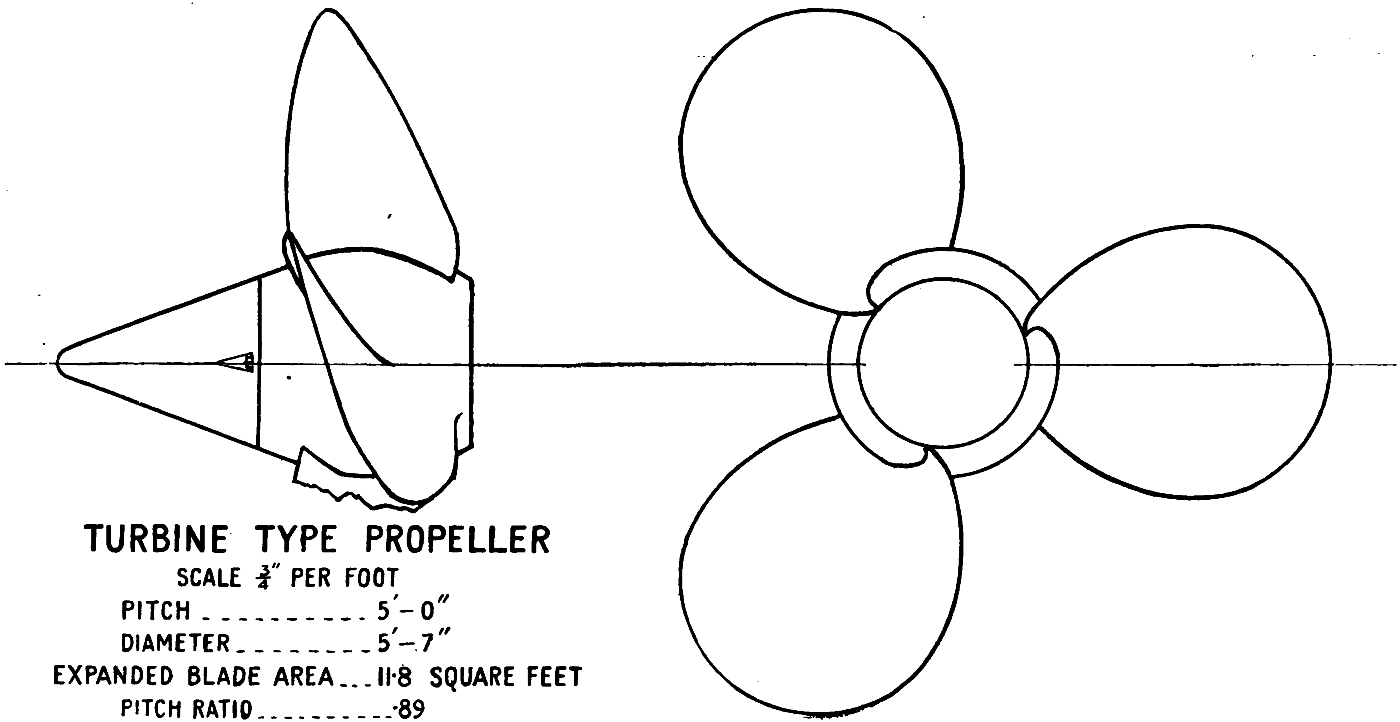
Propeller Efficiency.—Above a certain revolution speed a propeller of given diameter, pitch ratio, and area ratio rapidly loses in efficiency as cavitation sets in and reduces the effective thrust. The slip ratio therefore increases, which produces a correspondingly reduced propeller efficiency: it is thus impossible to obtain a maximum propeller efficiency in combination with a maximum turbine efficiency. Propellers of comparatively small diameter also possess the disadvantage of losing in efficiency against head seas and head winds, which again results in increased slip ratio. The problem is then to combine both propeller and turbine speeds so as to give the highest possible *combined* efficiency. It will have been observed that, particularly in turbine steamers, the actual sea speed falls away from the trial trip speed, and this will perhaps be understood from the foregoing explanation.

Cavitation.—Cavitation is caused by the ineffectiveness of the atmospheric pressure to press up the water at the back of the blades fast enough to allow of effective thrust. This usually occurs at high revolution speeds, and at high blade pressures per square inch.

Regarding cavitation and slip, Mr E. M. Speakman says:—

"Cavitation is partly the result of attempting to obtain too much work per square foot of blade area, and partly of excessive peripheral speeds. It has been found, by bitter experience occasionally, that there is a narrow limit to the tensional pressure possible on the water, per unit of projected area, beyond which the propeller efficiency drops very rapidly. This pressure is approximately from 10 to 12 lbs. per square inch at a depth of 12 inches below the surface, and to reduce the total thrust to this, sufficient blade area must be provided, which, in conjunction with certain practical proportions, necessitates a certain size of propeller, thereby limiting the revolutions.

"Friction and slip constitute the normal losses in all propellers, and augmented resistance must also be taken into account. This latter loss, however, is materially reduced with the smaller diameters of propeller found in turbine work. The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper,' down to about 14 per cent. in the 'Viking'—Channel steamers usually having



TURBINE TYPE PROPELLER

SCALE $\frac{3}{4}''$ PER FOOT

PITCH	5′–0″
DIAMETER	5′–7″
EXPANDED BLADE AREA	11·8 SQUARE FEET
PITCH RATIO	·89
AREA RATIO	·48

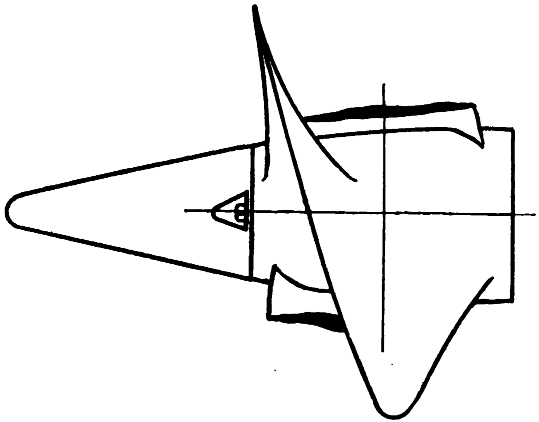
TURBINE PROPELLER.

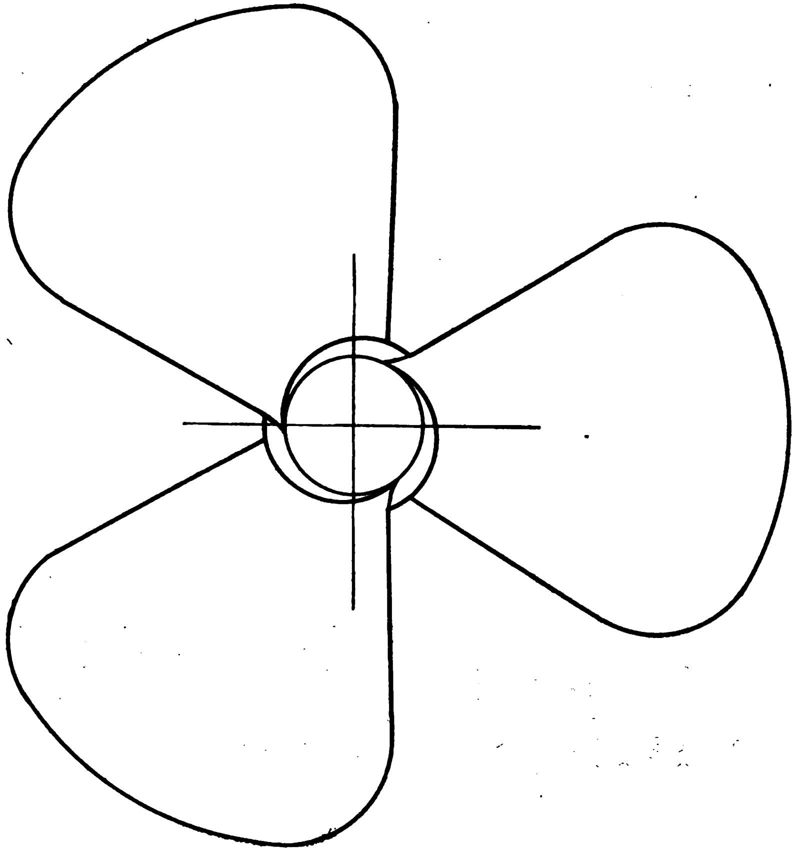
Pitch, 5 ft. ; Diameter, 6 ft. ; Pitch Ratio=.83.

PROPELLERS OF TURBINE STEAMER.

(Khedive's Yacht.)

Messrs A. & J. INGLIS & CO. LIMITED.





BRONZE TURBINE TYPE PROPELLER

DIAMETER 9′-3″ PITCH 8′-6″

PITCH RATIO ·91 AREA RATIO ·56

SCALE ½ INCH PER FOOT

about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Comparing the 'Manxman' with the 'Ulster'—a Holyhead mail boat of similar speed and power—the total disc area of the twin propellers of the latter is 226 sq. ft. (for two twelve-foot diameter propellers), while that of the 'Manxman' is only about 80 sq. ft. If the thrust deduction is proportional to the absolute area of the disturbance of the steam lines at the stern, the effect on the 'Ulster' will be far greater than on the 'Manxman,' but this action is to some extent affected by the intensity over the disturbing area which again is modified by the proximity of the propellers to the side of the vessel, this being less in turbine work. The disc area in H.M.S. 'Velox' is less than half that of the propellers in ordinary destroyers of the same power and speed. Cavitation is a preventable loss, and its presence on many vessels with insufficient blade area may be deduced from the falling off of the thrust curve and the rapid rise in the slip curve above a certain speed.

"From the analysis of numerous trials it appears that the pressure per square inch of projected area, when reduced to 12 in. immersion of tip, due to the effective thrust, is approximately 1 lb. for every 1,000 ft. per minute of circumferential velocity of blade tips. For a screw of a given pitch ratio, working at its maximum efficiency, this velocity should be proportional to the designed speed of the ship, and at full speed the pressures seem to have hitherto been about 5 lbs. per square inch for slow cargo vessels, from 6 to 7 lbs. for ocean-going mail steamers, and from 7.5 to 8.5 for cross-channel steamers; in cruisers and battleships they vary from 8 to 10.5 lbs. in some recent notable instances, in a torpedo craft from 9 to 11 lbs. At about 9 lbs., rather less possibly, the lowest suitable limit is reached for turbine screws; from 10 to 11 lbs. may be more usual in fast vessels, and though even from 12 to 14 lbs. has been known, pressures over about 11 lbs. always seem to be accompanied by low efficiencies. In large ocean-going vessels, which may be delayed by head winds and seas, a much lower designing pressure should be used, but in destroyers, as suggested above, something may be sacrificed at the maximum speed to obtain other advantages. The above pressure is worked out as mean pressure, as there exists no method of determining the local intensity per square inch, though the tendency of the distribution may be assumed in some cases. The

only published results are in Barnaby's papers on the trials of H.M.S. 'Daring,' but they are very inconclusive in many ways. Fig. 1 is submitted with much diffidence, merely as an illustration of the values of this limiting pressure.

"The maximum peripheral speed of tip ever used, I think, was 12,400 ft. per minute in H.M.S. 'Viper'; in H.M.S. 'Velox' it is about 11,650, and in the 'Londonderry' about 11,760, but many vessels have been below 9,000, which is quite normal for ordinary destroyer practice.

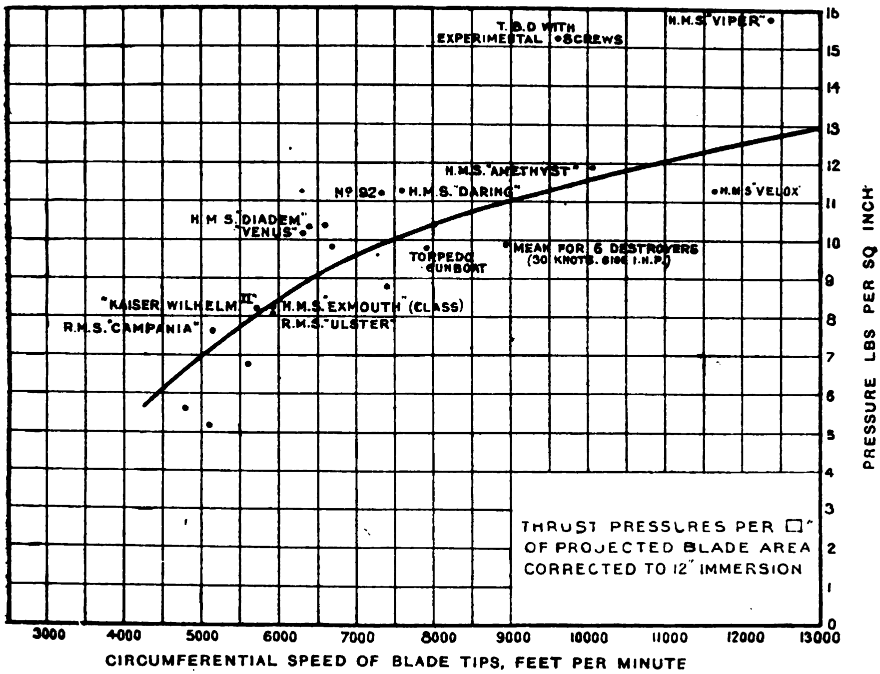


Fig. 1.

"The pitch ratio for turbine propellers has been purposely made considerably finer than usual. Thus, the pitch ratio for the 'Emerald' was about 0.6; in channel steamers and cruisers of from 18 to 25 knots it has varied from 0.8 to 1.0, and in torpedo craft from 1.0 in H.M.S. 'Velox,' to 1.35 in H.M.S. 'Viper,' and 1.6 in H.M.S. 'Cobra'; the latter vessels having 1, 2, and 3 screws per shaft respectively driven by identical turbines, the approximate revolutions at full speed being 900, 1,200, and 1,050 for 27, 36, and 31 knots respectively on trial.

"The percentage of slip has varied from 28 per cent. in H.M.S. 'Viper' down to about 14 per cent. in the 'Viking'—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Recent experiments in the model tank at Washington, D.C.,* seem to show that a symmetrical section will materially increase the pressure at which cavitation commences, and also demonstrate that in fine-pitch high-speed screws the back of the blade should receive almost as much attention as the face. This, as the author is well aware, is no new idea, but there have been repeated indications, especially in the trials of ordinary torpedo-boat destroyers, that while the gain may not be great, it is sufficient to merit attention. Mr Parsons has advocated a 10 per cent. reduction of pitch at the blade tip in order to avoid excessive local thrust, which might induce early cavitation, but there seems to be no advantage from departing from a true screw. The tendency of late years, in reciprocating-engine practice, has been to increase the ratio of projected to disc area from the .2 of Froude's classic screw, and the .22 to .26 of naval practice, o about .33; destroyer practice is included between this and .37, or even .4, at which point turbine practice may be said to commence. In this even from .5 to .56 has been used, but beyond, about .58 blade interference becomes excessive, and to obtain greater area a larger diameter must be used.

"The best form of blade is still undetermined. In the photo of the stern of the 'Lorena' will be seen the usual shape adopted, and experience seems to show that this almost circular shape, with the area disposed symmetrically on each side of the centre line, and with the generating line of the screw at right angles to the axis, gives as good results as any form.

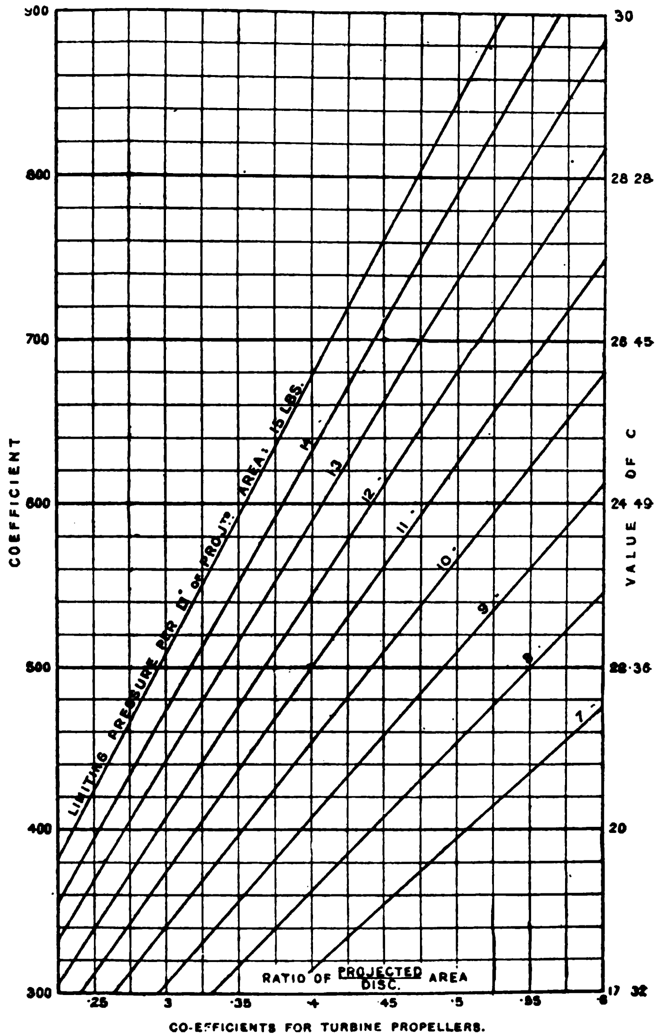
"I find that the following formula will give the diameter of a turbine propeller with considerable accuracy when the effective thrust along the shaft is known, and this must be calculated in any case if the steam balance of the turbine is to be good:—

$$\text{Diameter of propeller in feet} = \sqrt{\frac{\text{effective thrust in lbs.}}{\text{coefficient}}} = \frac{\sqrt{T}}{C}.$$

"This coefficient has been deduced from the limiting pressure per square inch, and the ratio of projected to disc area, and is

* See D. W. Taylor's paper, American Society N. A., 1904.

given in diagram form, in Fig. 2, where the full coefficient 400—900 is given in the left-hand scale, and values of C or



$$\text{DIA OF PROPELLER IN FEET} = \sqrt{\frac{\text{ACTUAL THRUST OF SCREW IN LBS.}}{\text{CO-EFFICIENT CORRESPONDING TO DESIRED PRESSURE AND AREA RATIO}}} = \sqrt{\frac{T}{C}}$$

Fig. 2.—Propeller Calculations.

$\sqrt{\text{Coefficient}}$ appear in the right-hand margin. The square root is only extracted for simplicity, whereby such coefficients as 30 are obtained for H.M.S. 'Viper'; while for the 'Manxman,' that for

the centre screw is 26.4, and for the wing screws about 28.75. For large ocean-going vessels with lower designing pressures these values will be rather less, perhaps about 22.

"The following table gives a few propeller dimensions and the corresponding coefficients, which the author trusts will be of use in designing high-speed screws:—

Propeller Dimensions.

Vessel.	Type.	No. of Screws.	Diameter.	Pitch.	Pitch Ratio.	Speed of Tip.	C Approximate.
			ft. in.	ft. in.		ft. p. min.	
Turbinia -	Experimental	9 3	1–6 2–4	2–0 2–4	1.33 1.0	10,860 ...	28 * 31.3*
Viper - -	T. B. D.	8	3–4	4–0 fwd 4–6 aft.	1.2 1.35	12,350	30.1
Amethyst -	3rd class cruiser	3^{1}_{2}	6–6	6– 6 5–10	1.0 .898	9,200 10,000	30.8
Manxman -	Cross-channel steamer	3^{1}_{2}	6–2 5–7	5–7 5–0	.906 .896	10,270 10,760	26.4 28.75
Londonderry	Cross-channel steamer	3^{1}_{2}	5–0	4–6	.9	10,550 11,810	30.8
Dieppe -	Cross-channel steamer	3	5–3	...	...	10,100 10,400	29
Carmania -	Atlantic mail	3	14–0	13–0	.928	8,125	21
Victorian -	Intermediate	3	8–30	...	...	7,150	24.65

* These C values are calculated from the same effective thrust in each case.

"Compared with above values of C, reciprocating-engine practice gives such figures as:—R.M.S. 'Lucania,' 16.5; H.M.S. 'Diadem' class, 17.5; H.M.S. 'Exmouth' class, 19.0; and standard 30-knot destroyers of 6,100 H.P., 20.8; all of which have much lower ratios of projected to disc area, and therefore a larger diameter and smaller C for a given power. Regarding these figures, it must be understood that C is an approximation only, and, owing to the difficulty of obtaining the actual power in each shaft in every case, cannot be considered as absolutely accurate. Probably, however, the error involved is under 2 per cent. It seems likely, to some at present undeterminable extent but within narrow limits for each class of vessel, that the propeller efficiency is proportional to the coefficient C, and this seems to be borne out by the trials of the 'Manxman' and the 'Londonderry.'

"Effective thrust is a somewhat subtle subject, and our know-

ledge of propulsive efficiency is by no means what it ought to be. These considerations will undoubtedly be brought into far greater prominence in the near future, and it is by no means improbable that the Admiralty or certain private owners will require some definite standard in this, just as coal or steam consumption is regulated at present. The more propeller efficiency is studied and understood the greater will be the improvement in the design of turbine installations for marine work; the turbine itself is a comparatively secondary consideration, and while at present propeller dimensions for turbine steamers can be quite as closely determined as those for ordinary work, the exact proportions must necessarily largely remain subject to modification from actual experience.

"While a general tendency has been very noticeable towards increasing the propeller diameter and reducing the revolutions, there will, of course, be some point, at present undetermined, at which the triple screws used in turbine work will be distinctly less efficient than ordinary twin screws. Very largely this is the case at present with triple screws driven by piston engines, on account of excessive thrust deduction and interference, but probably before this point is reached the weight of the turbines will have prevented its adoption.

"Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulæ exist at present, such as are met with in reciprocating-engine practice."

Under actual sea-going conditions, the resistance to advancement is increased by wind, &c., so that to obtain the same advance more revolutions are required, but as this increase in revolutions often produces cavitation (the propellers and revolutions being already designed for the limiting conditions), so that the limit is exceeded, and loss of push or thrust results, conjointly with a high slip ratio; hence, 15 per cent. slip may be the trial result, but 30 per cent. slip the actual sea-going result.

From the foregoing it will be evident that increasing the revolutions and decreasing the pitch may not give identical results, although theoretically this should be the case. The phenomenon of cavitation upsets the calculations, and seriously affects the results, owing to reduced thrust efficiency. Generally speaking—

1. A high turbine efficiency means a low propeller efficiency.

2. A high propeller efficiency means a low turbine efficiency.

The best combined efficiency of turbine and propeller is what has to be aimed at, and this can only be obtained by sacrificing one or other or both of the two efficiencies referred to, so that a compromise is effected. Generally, the propeller efficiency is sacrificed, as the advantage of this results in a higher proportional turbine efficiency and economy.

Turbine Efficiency.—The theoretical best efficiency of the turbine is attained when the linear velocity of the rotating blades is equal to about one-half the velocity of the steam impinging upon those blades, and as this figure is very high there remain but two alternatives to obtain the results which the best efficiency would require. The first of these two is to arrange the revolution speed so high as to enable the vanes to receive the steam under the conditions stated above. The other alternative is to reduce the speed of rotation of the turbine by increasing its diameter in equal ratio to the reduction of rotative speed, which, of course, has the disadvantage of increasing the weight of turbines required.

Turbine and Propeller Efficiency Combined.—It is often advisable in turbine steamers to sacrifice propeller efficiency so as to obtain a high turbine efficiency. This accounts for the high slip ratio noticeable in many turbine steamers, as it is found better to drop some of the propeller efficiency to gain more turbine efficiency.

Hence with high revolutions the turbine efficiency will be good, but the propeller may possibly give a better result with less revolutions per minute. A compromise is thus effected to produce the highest possible *combined* efficiency of turbines and propellers. This then explains the high slip per cent. often recorded.

Speed of Rotation.—As mentioned elsewhere, a great deal of the economy of a turbine motor depends on the high rotational velocity of the rotor, which, of course, necessitates smaller propellers, so that if the revolution

COMPARISON OF HIGH REVOLUTION AND LOW REVOLUTION SPEEDS ON TURBINE AND PROPELLER EFFICIENCIES.

HIGH REVOLUTIONS. (Say, 500 per min.)		LOW REVOLUTIONS. (Say, 150 per min.)	
Turbine Advantages.	Propeller Disadvantages.	Turbine Disadvantages.	Propeller Advantages.
1. Smaller turbines, therefore less weight of machinery.	1. Smaller propellers, therefore less holding power, or resistance against head winds and seas.	1. Larger turbines, therefore increased weight and space required for machinery.	1. Larger propellers giving increased blade surfaces, and more holding power.
2. Reduced blade clearance losses, as these vary inversely as revolutions squared.	2. Increased risk of cavitation due to increased resistance under severe sea-going conditions.	2. Largely increased blade clearance losses, as these vary inversely as revolutions squared.	2. Less pressure per square inch on blade surfaces, with correspondingly reduced risk of cavitation.
3. Better ratio of V.T. to V.S. to produce tur bine efficiency.	3. Reduced margin for increase of speed if required, as cavitation develops.	3. Great increase in centrifugal force due to increased weight of rotor. (The weight varies roughly as revolutions squared.)	3. Margin for increased speed without danger of cavitation developing.

speed is lowered to allow of larger propellers being used, the diameter of the turbine rotor must be increased if the same steam velocity is to be maintained: this means, in consequence, a rapid increase in weight of the rotors and casings.

From the foregoing it will be obvious that, roughly, the most advantageous conditions for the turbine constitute the most disadvantageous conditions for the propeller, and *vice versâ*, but as very elaborate and costly tank experiments have been made, and are still being carried on, the design of the propeller is being much improved, and the efficiency increased for higher revolution speeds. Experience seems to indicate that the propeller will require to meet the turbine requirements, and not the reverse.

Head Wind or Sea.—With small propellers, the effect of a strong head wind or current in retarding the ship's speed is more apparent, and constitutes an appreciable disadvantage; in consequence of this, large deep-sea turbine steamers are much lower in revolution speed than cross-channel or river turbine steamers, the propellers being larger in proportion.

Consumption and other Data.—Again referring to Mr E. M. Speakman's paper—

"A considerable amount of data on the performance of turbines compared with reciprocating engines for marine work is now available. The Admiralty has had tested both cruisers and torpedo-boat destroyers, exactly similar but for their engines and propellers, and trials of the Midland Railway Company's steamers and other cross-channel boats have corroborated the results regarding economy obtained from the naval vessels. In Fig. 3 is given the steam consumption per unit of power of H.M.S. 'Amethyst'* compared with that of several recent warships, and it is noticeable that only below from 55 to 60 per cent. of their full speed does the consumption of the turbine exceed that of the

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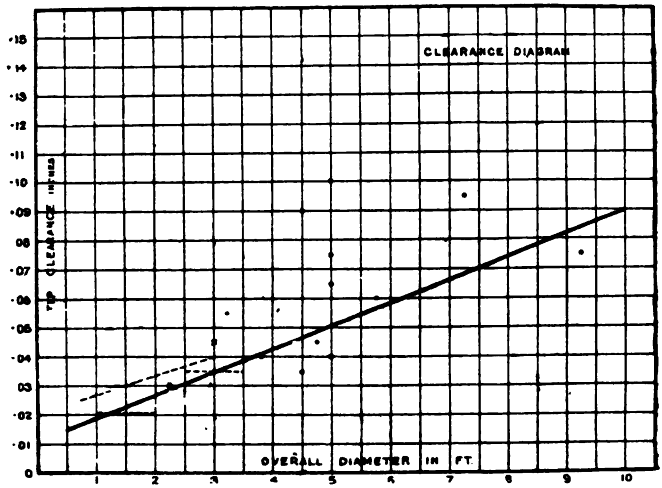
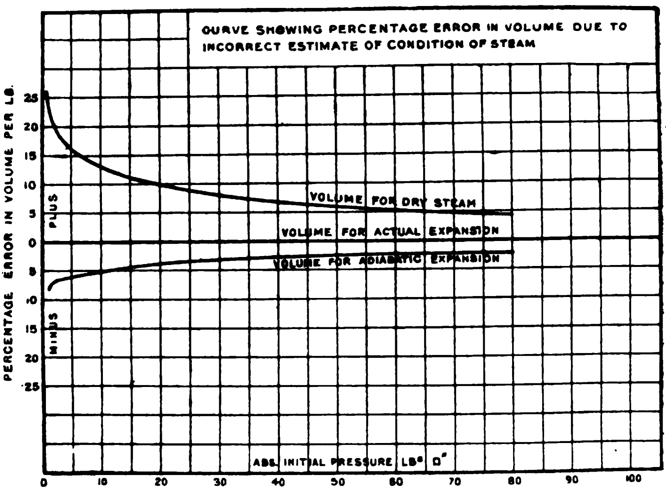
* Since the above-mentioned results were obtained, the steam piping has been altered so as to permit the auxiliary exhaust steam to pass through the main L.P. turbines when desired. This arrangement considerably decreases the consumption at low speeds, bringing the "Amethyst's" consumption below that of her sister ships down to 10 knots, or about 45 per cent. of full speed.

piston engines. Very seldom do vessels steam below these speeds. Cruisers carrying relief crews to the China or Australian stations usually proceed at about 60 per cent. of full speed, and in the



Tip Clearance Diagram.



Correction Factors for Condition of Steam.

NOTE.—These Diagrams refer to pages 38-41.

Atlantic manœuvres of 1903 nearly 80 per cent. of full speed was maintained by the large fleets, whilst the Japanese battleships built in England made their first voyage to Japan at about 54 per

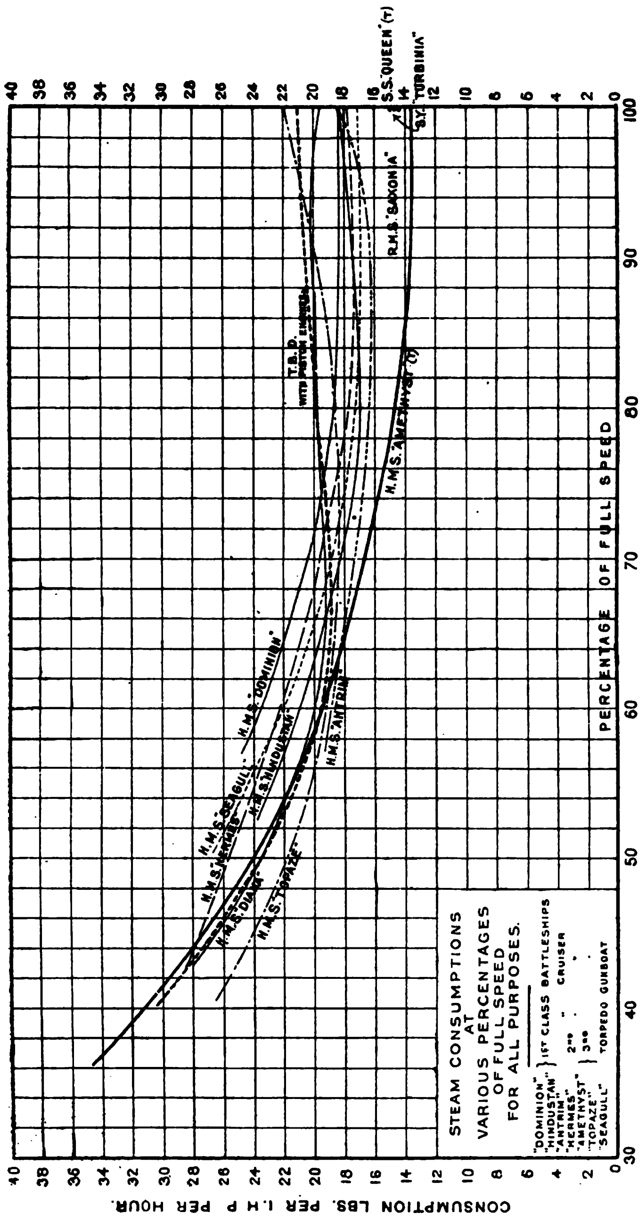


Fig. 3.—Steam Consumption per Hour for various Steamers.

cent. of their full speed; at which ratio the consumption per I.H.P. of both the 'Hindustan' and 'Dominion,' representing

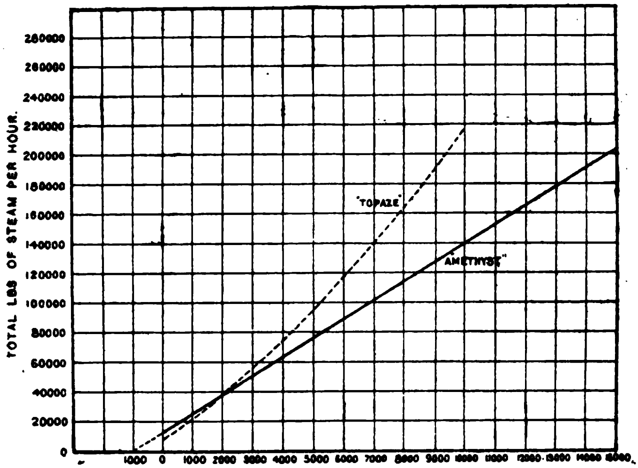


Fig. 4.—Steam Consumption.

H.M.S. "Topaze" (reciprocating). H.M.S. "Amethyst" (turbines).

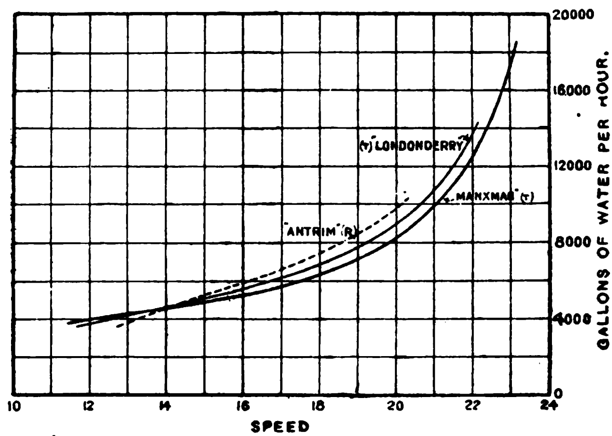


Fig. 5.—Water Consumption (Midland Railway Steamers).

SS. "Antrim" (reciprocating). SS. "Manxman" (turbines).
SS. "Londonderry" (turbines).

very recent battleship construction by eminent builders, is materially in excess of that of the first installation of warship turbines (not including the destroyers). The total consumption of H.M.S.

'Amethyst' and 'Topaze' plotted on a base of power is given in Fig. 4, while Fig. 5 shows that for the Midland Railway boats.* The progressive trials of H.M.S. 'Amethyst' are shown in Fig. 6, and, in view of the results obtained from these various vessels, the wholesale adoption of turbine machinery in the Royal Navy is not surprising.

"It is probable that the adoption of cruising turbines will be discontinued before long, and this view seems to be corroborated by the consumption trials of the Midland Railway steamers. Down to 60 per cent. of her full speed, the 'Manxman' required less water than the highly efficient 'Antrim,' and with a different blading arrangement in the main turbines, such a result should be equalled, if not improved on, in war vessels. The additional complication involved with two cruising turbines and their accompanying leakage and receiver losses, together with a considerable increase in weight and space occupied, largely modifies any advantages obtainable in the way of reduced consumption at lower powers. An improved (and easily obtainable) design of main turbine blading should give a better result at the highest powers, practically the same at intermediate powers (as in the case of H.M.S. 'Amethyst,' from 14 to 20 knots), and only slightly inferior at speeds below 14 knots, while it will undoubtedly admit of greater ease of handling and be much simpler. In a triple-shaft arrangement the unequal distribution of power on the wing shafts, due to the use of cruising turbines, is a distinct disadvantage: the fluctuation in rotative speed, due to shutting off the H.P. cruising turbine, may be seen from the trials of H.M.S. 'Amethyst.'"

Coal Consumption.—Regarding the all-important question of coal consumption, results have proved that at low or moderate speeds the reciprocating engine burns less per I.H.P. per hour than the turbine, but at high or maximum speeds the reverse is the case, the turbine showing in some cases an economy of as much as 20 per cent. and more over the reciprocating engine.

The difficulty in the way of a true comparison lies in the inability of determining the actual I.H.P. in turbine steamers, as the power cannot be thus indicated.

* Fig. 5 is compiled from the results given in Mr Grey's paper to the Institution of Naval Architects, July 1905.

Up to the present the deep-sea turbine steamers have not been able to compare very favourably with the ordinary engine in the matter of coal consumption, but for river or channel service the turbine steamers have run at a much lower consumption than those of the reciprocating type, as will be shown later.

The most conclusive tests as showing the superiority of the turbine over the triple engine at high speeds were those carried out some time ago by order of the Admiralty in the sister vessels "Topaze," "Sapphire," "Diamond," and "Amethyst," which were all designed and constructed similar in every particular, with the difference that the "Topaze," "Sapphire," and "Diamond" were supplied with triple-expansion engines, and the "Amethyst" with turbines.

The displacement of each vessel was 3,000 tons, and the estimated I.H.P. required for a speed of 21.7 knots was 9,800; it is therefore only reasonable and correct to assume that the power required to drive each vessel would be equal for the same speed. Working on this basis the results shown indicate a decided advantage in coal consumption in the turbine-propelled "Amethyst," as compared with the triple-expansion engined "Topaze" at the higher speed, but the reverse at lower speeds. The clear gain in coal at the maximum speeds is quite remarkable and constitutes a strong argument in favour of turbines; at 14 knots the conditions are, so far as economy is concerned, more equal; but when the speed was increased to 18 knots, it was found that the consumption on board the "Amethyst" was something like 20 per cent. less; at 20 knots it was nearly 30 per cent. less; and at the higher speed the improvement was still greater. The influence of this economy on the radius of action is very marked; for instance, the turbine-propelled ship could, with her 750 tons of coal on board, steam 3,160 sea-miles at 20 knots, as compared with 2,140 miles by the cruisers fitted with the ordinary machinery.

The water and coal consumption results of the turbine steamer "Amethyst" and triple-expansion engine "Topaze" are shown below.

WATER CONSUMPTION.

	"Amethyst." (Turbine.)	"Topaze." (Recip.)
24-Hours' Trial at 10 Knots.	(Turbine.)	(Recip.)
Indicated horse-power	897	897
Speed - knots	10	10.058
Total water per hour - lb.	26,260	21,294
Water per I.H.P. per hour - lb.	29.3	23.74
24-Hours' Trial at 14 Knots.		
Indicated horse-power	2250	2251
Speed - knots	14.062	14.08
Total water per hour - lb.	44,090	42,260
Water per I.H.P. per hour - lb.	19.6	18.77
30-Hours' Trial at 18 Knots.		
Indicated horse-power	4770	4776
Speed - knots	18.186	18.069
Total water per hour - lb.	76,493	90,500
Water per I.H.P. per hour - lb.	16	18.95
8-Hours' Trial at 20 Knots.		
Indicated horse-power	7280	6689
Speed - knots	20.6	20.063
Total water per hour - lb.	100,606	134,248
Water per I.H.P. per hour - lb.	13.8	20.07
4-Hours' Trial at Full Power.		
Indicated horse-power	13,000 14,000	9,573 9,868
Speed - knots	23.06 23.63	21.826 22.103
Total water per hour - lb.	176,845 190,525	209,950 199,140
Water per I.H.P. per hour - lb.	13.6 13.6	21.93 20.18

COAL CONSUMPTION.

		"Amethyst." (Turbine.)	"Topaze." (Recip.)
24 Hours' Trial at 10 Knots.		(Turbine.)	(Recip.)
Indicated horse-power		897	897
Total coal burnt	tons	31	24.6
Total burnt per hour	lb.	2893	2296
Total burnt per hour per I.H.P.	lb.	3.22	2.56
Evaporation per pound of coal	lb.	9.1	9.3
Miles run per ton of coal		7.42	9.75
24 Hours at 14 Knots.			
Indicated horse-power		2250	2251
Total burnt	tons	50.63	49.7
Total burnt per hour	lb.	4725	4640
Total burnt per I.H.P. per hour	lb.	2.1	2.06
Evaporation per pound of coal	lb.	9.35	9.13
Miles run per ton of coal		6.6	6.8
30 Hours at 18 Knots.			
Indicated horse-power		4770	4776
Total coal burnt	tons	112.13	146
Total burnt per hour	lb.	8372	10,900
Total burnt per hour per I.H.P.	lb.	1.75	2.28
Evaporation per pound of coal	lb.	9.15	8.3
Miles run per ton of coal		4.8	3.7
8 Hours at 20 Knots.			
Indicated horse-power		7280	6689
Total burnt	tons	39.06	55.2
Total burnt per hour	lb.	10,937	15,451
Total burnt per hour per I.H.P.	lb.	1.5	2.31
Evaporation per pound of coal	lb.	9.7	8.7
Miles run per ton of coal		4.22	2.9
4 Hours at Full Power.			
Indicated horse-power		13,000 14,000	9573 9868
Total coal burnt	tons	42.9 43.6	49.5 46.6
Total coal burnt per hour	lb.	24,035 24,412	27,700 26,130
Total burnt per hour per I.H.P.	lb.	1.85 1.74	2.89 2.65
Evaporation per pound of coal	lb.	7.35 7.8	7.56 7.95
Miles run per ton of coal		2.15 2.17	1.76 1.9

From the foregoing results it will be noticed that the consumption of coal per I.H.P. per hour at 18 knots is 1.75 lb. for the turbine and 2.28 lbs. for the triple-expansion engine, and at 20 knots the consumption of coal per I.H.P. per hour, 1.5 lb. for the turbine, and 2.31 lbs. for the triple-expansion engine, thus proving beyond question the economy of the turbine at high speeds.

Channel and river steamers show a decided gain in favour of turbines, the total consumption being actually less for greater speeds.

The following results, contained in a paper read by Mr William Grey, Member, at the Summer Meetings of the 46th Session of the Institution of Naval Architects, show the comparison very clearly.

"TABLE SHOWING RESULTS OBTAINED BY STEAMERS RUNNING SIMULTANEOUSLY, BUT IN OPPOSITE DIRECTIONS.

	Reciprocating Engine.	Turbine.
	'Antrim.'	'Londonderry.'
No. of trips - - - - - -	48	48
Average coal per trip (tons) - -	35.6	35.3
Average speed in knots - - -	19.7	19.5
	'Donegal.'	'Londonderry.'
No. of trips - - - - - -	42	42
Average coal per trip (tons) - -	36.0	36.9
Average speed in knots - - -	19.2	19.8
	'Antrim.'	'Manxman.'
No. of trips - - - - - -	29	29
Average coal per trip (tons) - -	38.6	38.6
Average speed in knots - - -	19.5	20.3
	'Donegal.'	'Manxman.'
No. of trips - - - - - -	39	39
Average coal per trip (tons) - -	38.7	40.2
Average speed in knots - - -	19.3	20.3

"These results point to a marked decrease in the coal consumption of the 'Manxman,' as compared with the 'Antrim and 'Donegal.' The 'Manxman' did 20.3 knots for the same coal consumption that the 'Antrim' had at 19.5 knots. A similar compari-

son of the 'Manxman' with the 'Donegal' gives nearly the same result.

"In other words, for a speed of 19.5 knots, the 'Antrim' requires 38.6 tons of coal and the 'Manxman' 35.0 tons, a saving of 9.3 per cent. The 'Donegal,' for a speed of 19.3 knots, requires 38.7 tons, and the 'Manxman' 35.4 tons, a saving of 8.5 per cent.

"The performances of the 'Londonderry' are nearly as efficient as those of the 'Antrim,' but they are better than those of the 'Donegal.'

"They also indicate that the 'Manxman,' with higher steam pressure, a smaller number of revolutions, and larger propellers, has done better than the 'Londonderry.'

"A further economy in the turbine steamers is effected in the amount of oil used for lubrication. The logs show that this amounts in both steamers to five gallons per single trip. This again permits of a further economy in the reduction of the engine-room staff from four greasers to two.

"Speaking generally, therefore, the performances of the turbine steamers, especially the 'Manxman,' have been greatly superior to those of the steamers fitted with reciprocating engines.

"It is not possible to make a quantitative analysis of the cost of upkeep, but so far the turbines have cost practically nothing (excepting the cost of repairs due to the accident to the 'Londonderry'), and they require very little attention compared with what is necessary in the very best running engines of the reciprocating type."

On the Clyde the turbine steamers "King Edward" and "Queen Alexandra" show marked improvement over steamers fitted with paddle engines, and it has been proved that in one case the "King Edward" steamed more distance in a given period than a paddle steamer of similar dimensions, on a consumption of 480 tons of coal *less*.

On the other hand, the ocean steamers "Victorian," "Virginian," and "Carmania," have not exactly come up to expectations in the matter of coal consumption, for reasons stated previously, and relating chiefly perhaps to propeller efficiency, or rather, loss of efficiency, under severe sea-going conditions; but, as also stated elsewhere, propeller design is undergoing constant improvement, and will, before long it is hoped, result in much reduced coal consumptions in turbine-propelled steamers of the Atlantic type.

A few examples of coal consumption, noted by the writer from actual practice, are appended.

(1.) I.H.P. 8,500, speed 22 knots, consumption about 6 tons per hour.

$$\text{Then, } \frac{6 \times 2240}{8500} = 1.58 \text{ lb. per hour per I.H.P.}$$

(2.) Shaft horse-power (not I.H.P.) 6,500, speed 20.7 knots, consumption about 5.2 tons per hour.

$$\text{Then, } \frac{5.2 \times 2240}{6500} = 1.79 \text{ lb. per hour per shaft H.P.}$$

(3.)

I.H.P. - - - -	9,000
Speed - - - -	22.8 knots.
Mean revolutions - -	520 per minute.
Boiler steam - - -	175 lbs.
H.P. turbine - - -	140 „
L.P. turbines - - -	12 „
Reverse turbines - -	22 in. vacuum.
Condenser - - -	27½ „ „
Consumption - - -	6.67 tons per hour.

$$\text{Coal per I.H.P. per hour} = \frac{6.67 \times 2240}{9000} = 1.66 \text{ lb.}$$

Running Astern.—In running astern, the consumption increases above that required for ahead, as the astern turbines are naturally not so efficient as those for ahead work.

Cruising Turbines.—In Admiralty work small additional cruising turbines have been fitted on two of the shafts. These turbines first receive the steam, which is afterwards expanded through the other turbines "in series," that is, successively. This system makes for economy at slow or medium speeds, but somewhat complicates the connections and general turbine arrangements. As the cruising turbines are of small diameter, the mean blade speed will be low, and the required steam speed will also require to be low if the usual ratio of V_t to V_s is to be maintained. This will necessitate a small pressure drop per

row and a resulting low power development. The economy really consists in the effective utilisation of the low-pressure steam as expanded from the cruising turbines into the ordinary ahead turbines by the "series" arrangement.

Power Developed by each Turbine.—(*A.*) Referring to Example 7, page 109, it will be seen that, running at a trial speed of 14.73 knots, the pressures indicated were as follows:—

H.P. turbine - - -	35 lbs. (gauge).
L.P. turbines - - -	12 in. vacuum.
Condenser - - - -	27 in. „

Assuming that the power varies as the cube of the speed, and as 9,000 horse-power gives 22 knots,

$$\text{Then, } \frac{14.73^3 \times 9000 \text{ I.H.P.}}{22^3} = 2701 \text{ Horse-Power.}$$

As there are three shafts, $2701 \div 3 = 900$ Horse-Power developed by each turbine; or, $900 \times 33{,}000 = 29{,}700{,}000$ foot-pounds of effective work per minute.

This energy is given out by each L.P. turbine in working between an initial pressure of 9 lbs. absolute and final pressure of about 2 lbs. absolute, or $9 - 2 = 7$ lbs. pressure drop in all, and this total pressure drop of 7 lbs. gives a heat drop sufficient to obtain 900 horse-power.

As, 29700000 foot-pounds $\div 778 = 38174$ Units Heat Drop per minute.

If each L.P. turbine is made up of say 64 rows of blades in all, then the mean pressure drop per row is equal to .10 of a pound, as $7 \div 64 = .10$; but the actual pressure drop is of a gradually decreasing quantity from the initial to the exhaust end of each turbine.

(*B.*) Again, taking the lowest trial speed of 10.68 knots, we find that the power required amounts to

$$\frac{10.68^3 \times 9000}{22^3} = 1030 \text{ Horse-Power.}$$

And as this is developed by the three shafts combined,

Then, $1030 \div 3 = 344$ Horse-Power per shaft or turbine,
and, $344 \times 33000 = 1032000$ foot-pounds of work per minute.
Therefore, $1032000 \div 778 = 1326$ units effective Heat Drop per minute.

The L.P. turbines are working in a vacuum of 21 in. at the initial end and 28 in. at the exhaust end. This heat drop is therefore given out by each L.P. turbine in working between pressures of 4½ lbs. initial and 1½ lb. final pressure (gross), or within a total pressure drop of $4\frac{1}{2} - 1\frac{1}{2} = 3$ lbs.

NOTE.—21 in. $\div 2 = 10.5$ lbs., and 15 lbs. atmospheric pressure $- 10.5 = 4.5$ lbs. absolute.

Therefore, $3 \div 64$ Rows $= .046$ of a pound pressure drop per Row.

The effective heat value of low-pressure steam as utilised in turbines will perhaps be apparent from the foregoing figures, which are taken from actual practice.

To sum up—

In case (*A*) the H.P. turbine develops 900 horse-power working between pressures of 50 lbs. absolute and 9 lbs. absolute. So that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $50 - 9 = 41$ lbs.

Each L.P. turbine also develops 900 horse-power, working *in a vacuum*, or between pressures of 9 lbs. absolute and 2 lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $9 - 2 = 7$ lbs.

In case (*B*) the H.P. turbine develops 344 horse-power working between pressures of 25 lbs. absolute and 4½ lbs. absolute, so that the pressure drop required for the necessary heat drop and foot-pounds of energy given out is equal to $25 - 4.5 = 20.5$ lbs.

Each L.P. turbine also develops about 344 horse-power, working *in a vacuum*, or between pressures of 4½ lbs. absolute and 1½ lb. absolute, so that the total pressure drop is 3 lbs., as $4.5 - 1.5 = 3$ lbs. The necessary heat drop is therefore obtained by this very small pressure drop, and the kinetic energy thus given up develops the required power.

DENNY & JOHNSON'S PATENT TORSION METER.

The following description of the appliance is reprinted from *Engineering*, 7th April 1905:—

The diagram, Fig. 1, shows the general arrangement of the apparatus as applied to turbine-driven shafting. On the shaft, the torsion of which it is desired to measure, are fixed, at a suitable distance apart, two light gun-metal wheels A and B. On each wheel is mounted, as shown, a permanent magnet, the projecting pole of which is made V shaped, in order to produce a dense and definite magnetic field at the point. Underneath the magnets and set concentrically with the wheels and shaft are fixed two inductors A and B, each of which consists of a quadrant-shaped piece of soft iron carried on a gun-metal stand provided with suitable levelling-screws. On each piece of iron are mounted a number of separate but similar windings of insulated wire, there being a certain suitable number of windings per unit of circumferential length of the iron. There is in conjunction with the inductors a recording box, in which are mounted two series, A and B, of contact studs, around which are fixed scales. In connection with the series of studs two contact arms A and B are arranged, by means of which electrical connection may be made at will between any desired stud of a series and its contact arm. There is in series A a stud for every separate winding in the inductor A, and in series B a stud for every separate winding in the inductor B, each stud being connected to its particular winding by means of a separate wire, all the wires being contained in the multiple cables A and B. The remaining ends or returns of the winding on the inductors A and B are all connected by means of two common wires (also contained in the cables A and B) to the contact arms A and B respectively.

Included in each of these two circuits is a variable resistance (by means of which the strength of the current flowing in the circuit may be adjusted as desired) and one winding of a differentially-wound telephone receiver. The scale A is divided into six equal parts, there being six separate windings in the inductor A, and thus six studs in the series A; the length of five subdivisions of the scale thus represents the circumferential length occupied by all the windings on the inductor, each subdivision

representing the distance between the neighbouring windings, which is usually 0.2 in. The scale B is divided into fourteen equal parts, there being fourteen separate windings on the inductor B, and thus fourteen studs in the series B. The length of thirteen subdivisions of the scale, as before, represents the circumferential length occupied by all the windings in the inductor, the distance between neighbouring windings being represented by one subdivision of the scale; the usual distance between neighbouring windings in inductor B is 0.02 in. The wheel in connection with

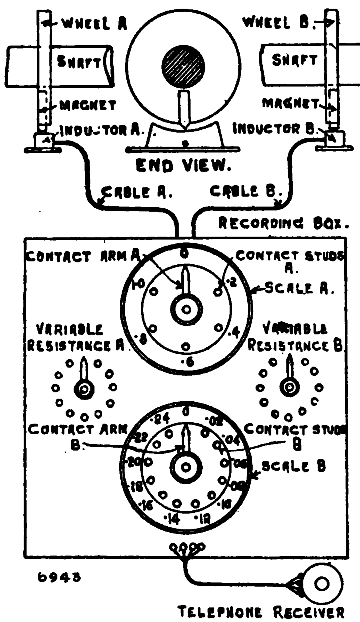


Fig. 1.

the inductor at the turbine end of the shaft is set so that the magnet fixed to it is exactly above one or other of the two end windings in the inductor.

The correct end winding to set the magnet to is that one from which the magnet will travel towards the other end winding when the shaft rotates. The wheel in connection with the other inductor is also set so that its magnet is exactly above one of the two end windings in the inductor, the correct end winding to set the magnet to in this case being that one from which the magn t will travel in the opposite direction to the other end winding when the

TORSION METER, SHAFT WHEEL, MAGNET, AND INDUCTOR.

shaft rotates. To facilitate the accurate and easy setting of the magnets above their respective windings, lines are cut in the tops of the inductors exactly above the end windings, and the magnets are set to these lines. When the shaft rotates without transmitting power, a current of electricity is induced in the end or zero winding of each inductor, the contact arms being first placed in contact with the end or zero stud in each series. These two separate currents both traverse their respective circuits, passing in each case from the inductor winding in which they are induced to the respective zero studs to which these windings are connected, thence by way of the respective contact arms, resistances, and telephone receiver windings back to the inductors again. The connections to the receiver windings are so arranged that the effects of the two separate currents flowing therein are in opposition, and thus neutralise each other's effect on the receiver when the strengths of the two currents flowing are exactly equal at the same instant. By means of the variable resistances in each of the circuits the currents are made equal in strength, and then so long as the shaft transmits no power, and is thus subject to no torsion, no sound will be heard on listening at the receiver, since the currents induced in the zero windings of the inductors have been equalised, and are both induced at exactly the same instant. When transmitting power, the shaft is subject to a certain torsion or twist, which causes the zero winding of the inductor next the turbine or engine to be excited in advance of the other by the amount of the torsion of the shaft; a loud ticking sound will then be heard in the receiver, as the currents no longer neutralise each other.

Contact arm B is then shifted from stud to stud, until the position of greatest silence in the receiver is once more obtained. When this position is found, the reading on the scale B, opposite the contact arm, represents the circumferential measurement of the angle of torsion of the shaft at the radius of the inductor windings. A current equal in strength at the same instant to that induced in the zero winding of inductor A is now being induced in that winding of inductor B which is in connection with the contact arm B; the scale reading thus represents the displacement of one magnet with regard to the other due to the torsion of the shaft. In the event of the torsion being found to be too great to be measured on scale B alone, contact arm A is shifted from stud to stud until a reading can be obtained on scale B, the torsion reading being equal to the sum of the readings on scales A and B. The reading corresponding to any large displacement of one

magnet relatively to the other is thus easily obtained by the combined use of the scales A and B.

TYPE A INSTRUMENTS.

This type has been designed for measuring the torsion of shafts having uniform turning moments, *e.g.*, shafting driven by turbines, electric motors, &c. &c.

To Fix the Apparatus.—A set of instruments as applied to a shaft consist of the following items:—(1) Two light gun-metal wheels, each fitted with a permanent magnet; (2) two inductors; and (3) a recording box. Before fixing up the apparatus it is first necessary to determine what length of the available shafting will be required to give a reasonably large torsional reading. The power to be transmitted by, and the revolutions of, the shaft at the maximum load are first approximately calculated. The diameter of the shaft is then measured, and the inductor constant noted. (This constant is the actual radius in inches at which the torsion of the shaft is measured; and it is stamped on all inductors.) Having made these observations, the length of shafting in feet required to give approximately 1 in. of reading is obtained by solving the formula,

$$\frac{1.53 \times R \times d^4}{C \times H.P.} = \text{Length}.$$

Where R = revolutions per minute of the shaft, C = inductor constant, d = diameter of shaft in inches, H.P. = horse-power (approximate), and 1.53 is a constant which takes account of the figure 140, a constant which has been ascertained from a great number of static calibrations of shafts, and which is an expression of the torsional resistance of solid iron and steel shafting. In the case of hollow shafting the torsional strength can best be found by calibrating the shaft (see page 148), but it is probable that the torsional strength is the same as that of a solid shaft of the same diameter minus the resistance due to a solid shaft which would fill the bore, and, in the absence of calibration, the error due to this assumption is probably a negligible quantity.

Having found the length of shafting which will give 1 in. of reading, this length is selected if available, and the wheels fixed to the shaft. It is desirable to select the greatest length of shafting available, provided the length chosen is such that the full load reading shall not exceed 1 in., it being advisable to have the remainder of the scale in reserve in the event of the power, &c.,

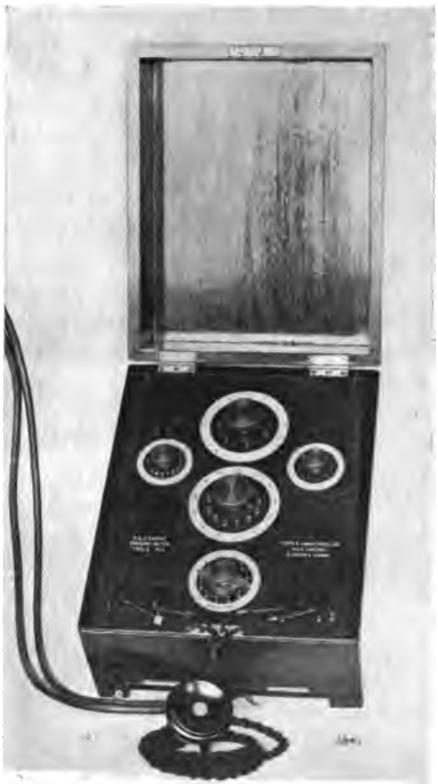
having been underestimated. In most cases it will be advisable to sacrifice a short length of the available shafting in order that the wheels may be fixed close to bearings (as shown in the illustration), thus allowing the wood or iron soles on which the inductors are to be mounted to be bolted to the bearing stools. The advantage of this is that any vibration that may be present when the shaft is running affects the shaft and bearing stools equally. If necessary, however, the inductors may be fixed in any other position, provided that a good solid base is provided. After fixing the wheels in place, the inductors should be screwed to the soles, and set concentrically with the wheels. The setting is done by means of two setting pieces supplied with the instrument. These are inserted between the top of the inductor and its wheel, the levelling screws in the base of the inductor being adjusted until the setting pieces both fit tightly and bear throughout their entire length. The locking nuts in the base are then tightened up and the setting pieces removed. The top or covering plate of each inductor has two lines or grooves cut across it, and the wheel in connection with the inductor fixed at the turbine or engine end of the shaft must be adjusted on the shaft until the magnet on that wheel is directly above the line from which it will travel from, and towards the other line when the shaft rotates. *It is necessary to set the magnet above the correct line (which is the zero line corresponding to the direction of rotation of the shaft) or no readings can be taken.* The magnet is arranged in a tight fitting groove in the wheel, so that by removing the bridge pieces which hold it in position the magnet may be lowered until its sharp edge rests exactly in the zero line. The wheel should then be clamped tight to the shaft, and the magnet raised and fixed in position with its edge about $\frac{1}{16}$ in. clear of the top of the inductor. A gauge-marked air gap is supplied for the purpose of testing the clearance.

In addition to the two zero lines there is also a line marked circumferentially on the top of the inductor, and when setting the inductor this line should be made to coincide with the lines which are cut on the magnet. The other inductor is set and fixed in a similar manner; but in this case the magnet must be set to the zero line at the opposite end of this inductor. The inductors must be so placed that the connections are on the same side, *i.e.*, both on the right or both on the left hand side of the inductors, in order that the induced currents of electricity may be in the proper directions, and the trouble of altering the telephone connections thus avoided. The cables should now be fixed to the

inductors. The plugs attached to the ends of the cables are inserted into the sockets on the inductors, care being taken to see that the numbers on the plug and socket carriers correspond. The plugs must all be so inserted that the arrows on the plug carriers always point from the lines to which the magnets are set and in the direction of the other line. (This is essential.) The cables are not the same length. The longer one is intended to go to the inductor farthest away from the recording box (care should therefore be exerted when fixing the inductors to see that the inductor intended for use with the long cable is placed at the proper end of the shaft). This will probably be the after-end in all ships. The recording box should be placed in a reasonably quiet place, preferably a room as far removed from the noise of the machinery as is possible. The cable connections are then made to the recording box, and as the plugs and sockets are numbered no mistake should be possible.

To Take a Reading.—First see that the *resistance* contact arms and the *main* contact arms in the recording box are both at zero. The telephone receiver is then placed to the ear of the observer, and when the shaft is revolving a ticking sound will be heard. Shift the *main* contact arm of the $\frac{1}{100}$ths scale from stud to stud until a position is found where no tick is heard, or where the sound is reduced to a minimum. The torsion reading will then be found on the scale opposite the *main* contact arm. Should the tick not cease entirely but appear to be equally low on each of two neighbouring studs, the correct position of the arm is between those studs. When no reduction of the ticking noise can be obtained by moving this arm, it is evident that the torsion of the shaft is greater than the range of the $\frac{1}{100}$ths scale. The arm of the $\frac{1}{5}$ths scale must then be shifted from stud to stud until a reading can be obtained on the $\frac{1}{100}$ths scale. The torsion reading will then be the sum of the two scale readings. We have assumed that no balancing or equalising of the currents by resistance was required, but in practice this will usually require to be done before readings can be accurately taken. The equalising is accomplished as follows:—First take a trial contact arms reading as described above, and then move one of the resistance (the correct one being found by trial) until a position is found where the sound in the receiver almost or entirely disappears. It may be necessary to slightly alter the trial reading, as well as the resistance, before silence is obtained, and in cases where the equalising cannot be satisfactorily accomplished this will no doubt

be due to the receiver being too sensitive for the rate of revolution. This can be remedied by adjusting the tension screw at the back of the telephone receiver until the desired degree of sensitiveness is obtained. Having once found the correct position for the resistance contact arm, no further adjustment will be



Recording Box.

necessary for a particular shaft or set of shafts. When applying the instrument to another shaft, or in cases where the revolutions of the shaft under trial vary greatly, it may be desirable to slightly readjust the resistance contact arm and also the receiver. With turbine-driven shafting in ships, where the rate of revolution is sufficiently high, a very good check may be applied to the accuracy

of the zero setting of the inductors in the following manner:—After the day's trial is over, let the steam be shut off one turbine, and while the ship is driving ahead with the remaining turbines note the reading of the idle shaft, which will continue to revolve due to the action of the water on its propeller. The friction of the shaft bearings and of the turbine may be assumed to require a relatively negligible amount of power to be transmitted by the propeller along the shaft, so that the reading thus taken should agree with the zero reading. To find the zero reading by this check method, the main contact arms of the two scales are placed opposite the letters Z, when no sound should then be heard in the receiver if the inductors have been properly set. If a sound be heard, the arm of the $\frac{1}{100}$ths scale must be shifted until the tick disappears, or until the sound is reduced to a minimum. If the reading thus obtained is less than the reading at Z, then the difference between the two readings is the amount of zero error, and must be added to the whole of the readings previously taken. If, on the other hand, the reading is greater than that at Z, then the difference between the two readings must be subtracted from the whole of the readings previously taken. The foregoing test need only be applied, however, in the event of doubtful readings being obtained, as the set zero will always be correct if the inductors are carefully set up.

To Calibrate the Shaft.—Previous to being placed in its position the shaft should be calibrated. This is done by coupling the various lengths together to include the length over which the torsion is to be measured, and then setting it on bearers and twisting it by means of weights in the following manner:—Clamp one end of the shafting rigidly in position so that it cannot turn, and fix a stout lever to the other end. On to the same end of the shaft fix a pointer 6 or 7 ft. long, so that it will move over a paper scale. Weights should then be attached to the outer end of the lever, and the deflection of the pointer in inches noted as the weight is increased until a set of readings from zero up to the maximum calculated torque are obtained. A curve is then drawn, the ordinates being foot-pounds and the abscissal torsion in inches.

To Find the Shaft H.P. from the Torsion Reading.—From the calibration curve the foot-pounds corresponding to any torsional reading of the shaft taken by the torsion meter may be easily found. The torsion meter readings must first of course be

made to correspond with the length of calibrated shaft and to the calibration radius by means of the formula

$$\frac{r \times l \times L^1}{C \times L} = K$$

where r = torsion meter reading in inches, l = length of pointer in inches, L^1 = length in feet of calibrated shaft, C = inductor constant, and L = length in feet of shaft from which the torsion meter reading is obtained, K = reading of torsion meter corresponding to the curve.

Having found from the curve the foot-pounds corresponding to any reading of the torsion meter, the horse-power transmitted is then found by means of the formula

$$\frac{F \times R}{5255} = \text{horse-power transmitted by the shaft,}$$

where F = foot-pounds, R = revolutions per minute of the shaft, and 5255 is a constant.

In the case of solid shafts of iron or steel, instead of calibrating the shaft the constant 140 (which figure represents the average torsional resistance of iron and steel shafts as found from numerous calibrations of shafts) may be accepted as correct, and the horse-power transmitted by the shaft found by means of the formula

$$\frac{1.53 \times r \times d^4 \times R}{C \times L} = \text{horse-power.}$$

In the above formula, d = diameter in inches of the shaft, R = revolutions per minute of the shaft, C = inductor constant, L = length in feet of shaft, r = reading of torsion meter in inches, and 1.53 is a constant.

Hollow shafting should be calibrated, as the torsional resistance varies with the bore, which may not be sufficiently uniform to permit of very accurate calculation.

It is strongly recommended, however, that all shafting, whether hollow or solid, should be calibrated if at all possible, as very great accuracy is then assured. When the constant 140 is used in place of calibration the average possible error is about 1 per cent.

ELEMENTARY PROBLEMS IN MARINE TURBINE DESIGN.

(For Junior Students.)

Steam and Blade Speeds.

NOTE.—In marine practice, the ratio of V_t to V_s varies from about .37 to .48,

Where V_t = peripheral blade speed.
" V_s = steam speed.

1. Calculate the mean blade velocity if the steam speed is 300 feet per second, and the ratio $V_t \div V_s = .4$.

Answer. 120 feet per second.

2. What is the necessary steam speed per second for a blade velocity of 90 feet per second, if the ratio $V_t \div V_s$ is equal to .45?

Answer. 200 feet per second.

3. The ratio of V_t to V_s is .48, and the steam speed 250 feet per second. Find the mean blade velocity.

Answer. 120 feet per second.

4. If the steam speed is 350 feet per second, and the blade velocity 150 feet per second, calculate the ratio of V_t to V_s.

Answer. .42 ratio.

Diameter of Rotors, &c.

RULE.—Blade Velocity per minute = Rotor diameter × 3.1416 × Revolutions.

Therefore, $\dfrac{\text{Blade Velocity}}{3.1416 \text{ Revolutions}}$ = diameter of Rotor (across blades),

and, $\dfrac{\text{Blade Velocity}}{\text{Diameter} \times 3.1416}$ = Revolutions.

NOTE.—The mean blade Velocity varies from .3 to .5 of the steam Velocity.

1. Calculate the required diameter of L.P. rotor for a mean blade velocity of 9,000 feet per minute, and revolutions of 400 per minute.

Answer. 7.16 feet diameter.

2. Calculate the required diameter of H.P. rotor if the mean blade velocity is 6,000 feet per minute, and revolutions 550 per minute. Answer. 3.47 feet diameter.

3. Calculate the required diameter of reverse rotor for a mean blade velocity of 5,000 feet per minute, and revolutions of 400 per minute. Answer. 3.97 feet diameter.

4. Determine the required revolutions for a turbine if the mean blade velocity is 8,000 feet per minute, and rotor diameter 6 feet. Answer. 424 revolutions.

5. Find the revolutions to correspond with a blade velocity of 9,000 feet per minute, and rotor diameter of 7 feet.
Answer. 409 revolutions.

6. Calculate the necessary revolutions for a blade velocity of 100 feet per second, and rotor diameter of 5 feet 6 inches.
Answer. 347 revolutions.

7. Find the revolutions for a blade velocity of 90 feet per second, and rotor diameter of 4 feet.
Answer. 429 revolutions.

8. Find the revolutions required for a blade velocity of 160 feet per second, and rotor diameter of 14 feet.
Answer. 218 revolutions.

Steam Velocities and Heat Drops.

NOTE.—British thermal units $\times$ 778 = Foot-pounds of energy given up.

And foot-pounds $\div$ 778 = B.T.U. Heat Drop.

Again, $\dfrac{W \times V^2}{64.4}$ = foot-pounds of energy in first guide blades.

NOTE.—W = Weight in pounds.
V = Velocity of steam in feet per second.

Also, $\dfrac{(V^2 - v^2) \times W}{64.4}$ = foot-pounds given up in any moving blades.

Therefore, $64.4 \times$ foot-pounds $= W \times V^2$.

And, $\sqrt{\dfrac{64.4 \times \text{foot-pounds}}{W}} = V$ in first guide blades.

Or, $\sqrt{\dfrac{64.4 \times \text{foot-pounds} + v^2}{W}} = V$ in moving blades.

NOTE.—V = Velocity of steam in feet per second at exit edge of blades.

v = Velocity of steam in feet per second at admission edge of blades.

W = Weight of steam in pounds.

64.4 = Gravity's acceleration, or 32.2 feet per second, per second.

1. Calculate the work done on one pound of steam in passing through the guide blades of a Parsons turbine at an initial velocity of 600 feet per second. Answer. 5,590 foot-pounds.

2. A pound of steam enters the guide blades of a marine turbine at a velocity of 300 feet per second. Calculate the work done and the heat given up.

Answer. 1,397 foot-pounds; 1.79 B.T.U.

3. Calculate the foot-pounds of kinetic energy done in one pound of steam when passing the first guide blades of a turbine at a velocity of 350 feet per second, also the B.T.U.

Answer. 1,902 foot-pounds; 2.44 B.T.U.

4. Calculate the foot-pounds of energy and drop of heat which are produced by one pound of steam at a velocity of 15,000 feet per minute. Answer. 970 foot-pounds; 1.24 B.T.U.

5. One pound of steam enters the moving blades of a marine turbine at a velocity of 200 feet per second, and leaves at a velocity of 300 feet per second. Calculate the foot-pounds of energy given out and the heat drop.

Answer. 776 foot-pounds; .99 B.T.U.

6. Find the heat drop if the steam velocities are 300 feet and 400 feet per second. Answer. 1.39 B.T.U.

7. Find the required heat drop to give velocities of 300 feet and 220 feet per second in the moving blades.

Answer. .82 B.T.U.

8. Calculate the steam velocity per second if the energy given out is 2,000 foot-pounds for each pound of steam supplied.

Answer. 358 feet per second.

9. Find what velocity of steam is necessary to develop 1,000 foot-pounds of energy per pound of steam.

Answer. 253 feet per second.

10. Twenty pounds of steam passing through the guide blades of a turbine give out 10,000 foot-pounds of work. Calculate the velocity of the steam. Answer. 179 feet per second.

11. One pound of steam drops 2 B.T.U. in passing through the blades of a turbine. Calculate the steam velocity.

Answer. 316 feet per second.

12. What steam velocity is required to obtain a heat drop of 2.5 B.T.U. per pound of steam?

Answer. 353 feet per second.

13. Calculate the exit velocity of the steam in the turbine blades if the foot-pounds given out per pound are 1,500, and the initial velocity is 200 feet per second.

Answer. 369 feet per second.

14. The initial velocity is 300 feet per second, and the heat units given up 2. Calculate the steam velocity at the exhaust edge. Answer. 436 feet per second.

15. What exit velocity is necessary if the heat drop is 1.5 B.T.U., and the initial velocity 260 feet per second?

Answer. 377 feet per second.

16. Heat drop 3 B.T.U., initial velocity 6,000 feet per minute. Calculate the exit velocity. Answer. 400 feet per second.

17. One pound of steam contains 1,200 B.T.U. on entering the blades, and 1,196 B.T.U. on leaving the blades of a turbine. Calculate the exit velocity if the admission velocity is 300 feet per second. Answer. 538 feet per second.

18. Calculate the exit velocity of the steam if the initial velocity is 160 feet per second and the heat drop 1.75 B.T.U.

Answer. 336 feet per second.

19. Initial velocity of steam 200 feet per second, and heat drop 1.5 unit. Calculate the exit velocity.

Answer. 339 feet per second.

20. In a marine turbine one pound of steam at an initial velocity of 200 feet per second passes through the moving blades. Find the final velocity if the heat drop is 1.8 B.T.U.

Answer. 360 feet per second.

Calculations for Blade Velocities and Number of Rows.

RULE.—

Blade Velocity per second $^2 \times$ Number of Rows = Constant.

Therefore, Constant $\div$ Blade Velocity ^{2} = Number of Rows,

and, $\sqrt{\text{Constant} \div \text{Number of Rows}}$ = Blade Velocity.

NOTE.—The Constant referred to varies from 1,400,000 to 1,600,000 (see page 38).

1. Calculate the required total number of rows of ahead blades required for a Parsons marine turbine if the constant is 1,400,000 and the blade velocity 100 feet per second.

Answer. 140 rows of blades.

2. Determine the total number of ahead blade rows required for the three rotors of a turbine if the blade velocity is to be 80 feet per second and the constant allowed is 1,500,000.

Answer. 150 rows of blades.

3. Find the suitable mean blade velocity if the number of rows is 192 and the constant 1,600,000.

Answer. 91.2 feet per second.

4. What blade velocity per second will be necessary if the ahead turbine rotors contain in all 168 rows of blades and the given constant is 1,500,000? Answer. 94.4 feet per second.

5. The ahead H.P. and L.P. turbine rotors each contain 48 rows of blades. Calculate the required blade velocity if the constant is 1,500,000. Answer. 102 feet per second.

6. Determine the number of blade rows required in each ahead rotor if the mean blade velocity is 120 feet per second and the constant 1,600,000.

Answer. 111 rows in all; 37 rows in each rotor.

7. Determine the constant used when the blade velocity is 90 feet per second and the number of rows 144.

Answer. 1,166,400 constant.

8. The number of ahead blade rows is 156 and the mean blade velocity 100 feet per second. Calculate the constant which has been employed. Answer. 1,560,000 constant.

Turbine Propeller Calculations.

RULE.—

$$\sqrt{\text{Effective Thrust on Shaft}} = C \times \text{Diameter of Propeller.}$$

$$\text{Therefore, } \frac{\sqrt{\text{Effective Thrust}}}{C} = \text{Diameter of Propeller.}$$

NOTE.—C = Constant found from suitable coefficient for blade area ratio and blade pressure per square inch (see page 124).

1. Determine the diameter of turbine propeller suitable if the effective thrust on each shaft is 23,180 lbs. and the constant 29.
Answer. 5.24 feet diameter.

2. Calculate the diameter of turbine propeller required when the thrust per shaft is 24,000 lbs. and the constant C 30.
Answer. 5.15 feet diameter.

3. Find the required diameter of propeller if the effective thrust is 60,000 lbs. on each shaft and the constant 24.63.
Answer. 9.94 feet diameter.

4. Calculate the propeller diameter if the effective thrust in each shaft is found to be 44,000 lbs. and the constant 29.
Answer. 7.23 feet diameter.

5. What diameter of propeller should be employed when the thrust is 30,000 lbs. on the shaft and the constant is 30?
Answer. 5.77 feet diameter.

6. Determine the effective thrust pounds if the propeller diameter is 5 feet and the constant 31.
Answer. 24,025 lbs. thrust on each shaft.

7. What is the effective thrust in pounds if the propeller diameter is 14 feet and the constant 21?
Answer. 86,436 lbs. thrust on each shaft.

8. What diameter of propeller should be used when the effective thrust is 25,000 lbs. and the constant is 28?
Answer. 5.64 feet diameter.

9. What diameter of propeller should be fitted to each of three shafts when the effective thrust is 91,476 lbs. and the constant is 21? Answer. 14.3 feet diameter.

10. What diameter of propeller should be used when the effective thrust is 188,051 lbs. and the constant 25, the steamer having 4 lengths of shafting? Answer. 17.3 feet diameter.

SOLUTIONS TO PROBLEMS IN MARINE TURBINE DESIGN.

Steam and Blade Speeds.

1. $300 \times .4 = 120$ feet per second. Answer.

2. $90 \div .45 = 200$ feet per second. Answer.

3. $250 \times .48 = 120$ feet per second. Answer.

4. $150 \div 350 = .42$. Answer.

Diameter of Rotors, &c.

1. $\frac{9000}{3.1416 \times 400} = 7.16$ feet Diameter. Answer.

2. $\frac{6000}{3.1416 \times 550} = 3.47$ feet Diameter. Answer.

3. $\frac{5000}{3.1416 \times 400} = 3.97$ feet Diameter. Answer.

4. $\frac{8000}{6 \times 3.1416} = 424$ Revolutions. Answer.

5. $\frac{9000}{7 \times 3.1416} = 409$ Revolutions. Answer.

6. $\frac{100 \times 60}{5.5 \times 3.1416} = 347$ Revolutions. Answer.

NOTE.—Multiply by 60 seconds for one minute.

7. $\frac{90 \times 60}{4 \times 3.1416} = 429$ Revolutions. Answer.

8. $\frac{160 \times 60}{14 \times 3.1416} = 218$ Revolutions. Answer.

Steam Velocities and Heat Drops.

1. $\frac{1 \times 600^2}{64.4} = 5590$ foot-pounds. Answer.

2. $\frac{1 \times 300^2}{64.4} = 1397$ foot-pounds. Answer.

   And, $1397 \div 778 = 1.79$ B.T.U. Answer.

3. $\frac{1 \times 350^2}{64.4} = 1902$ foot-pounds. Answer.

   And, $1902 \div 778 = 2.44$ B.T.U. Answer.

4. $15000 \div 60 = 250$ feet per second.

Then, $\frac{1 \times 250^2}{64.4} = 970$ foot-pounds. Answer.

And, $970 \div 778 = 1.24$ B.T.U. Answer.

5. $\frac{(300^2 - 200^2) \times 1}{64.4} = 776$ foot-pounds. Answer.

And, $776 \div 778 = .99$ B.T.U. Answer.

6. $\frac{(400^2 - 300^2) \times 1}{64.4} = 1086.95$ foot-pounds,

And, $1086.95 \div 778 = 1.39$ B.T.U. Answer.

7. $\frac{(300^2 - 220^2) \times 1}{64.4} = 645.9$ foot-pounds,

And, $645.9 \div 778 = .82$ B.T.U. Answer.

8. $\sqrt{\frac{64.4 \times 2000}{1}} = 358$ feet per second. Answer.

9. $\sqrt{\frac{64.4 \times 1000}{1}} = 253$ feet per second. Answer.

10. $\sqrt{\frac{64.4 \times 1000}{20}} = 179$ feet per second. Answer.

11. $\sqrt{\frac{64.4 \times 2 \times 778}{1}} = 316$ feet per second. Answer.

12. $\sqrt{\frac{64.4 \times 2.5 \times 778}{1}} = 353$ feet per second. Answer.

13. $\sqrt{\frac{64.4 \times 1500 + 200^2}{1}} = 369$ feet per second. Answer.

14. $\sqrt{\frac{64.4 \times 2 \times 778 + 300^2}{1}} = 436$ feet per second. Answer.

15. $\sqrt{\frac{64.4 \times 1.5 \times 778 + 260^2}{1}} = 377$ feet per second. Answer.

16. $6000 \div 60 = 100$ feet per second.

$\sqrt{\frac{64.4 \times 3 \times 778 + 100^2}{1}} = 400$ feet per second. Answer.

17. $1200 - 1196 = 4$ B.T.U. Drop.

$\sqrt{\frac{64.4 \times 4 \times 778 + 300^2}{1}} = 538$ feet per second. Answer.

18. $\sqrt{\frac{64.4 \times 1.75 \times 778 + 160^2}{1}} = 336$ feet per second. Answer.

19. $\sqrt{\dfrac{64.4 \times 1.5 \times 778 + 200^2}{1}} = 339$ feet per second. Answer.

20. $\sqrt{\dfrac{64.4 \times 1.8 \times 778 + 200^2}{1}} = 360$ feet per second. Answer.

Calculations for Blade Velocities and Number of Rows.

1. $1400000 \div 100^2 = 140$ Rows of blades. Answer.
2. $1500000 \div 100^2 = 150$ Rows of blades. Answer.
3. $\sqrt{1600000 \div 192} = 91.2$ feet per second. Answer.
4. $\sqrt{1500000 \div 168} = 94.4$ feet per second. Answer.
5. $\sqrt{1500000 \div 48 \times 3} = 102$ feet per second. Answer.
6. $1600000 \div 120^2 = 111$ Rows in all, or 37 Rows in each rotor. Answer.
7. $90^2 \times 144 = 1166400$ Constant. Answer.
8. $100^2 \times 156 = 1560000$ Constant. Answer.

Turbine Propeller Calculations.

1. $\dfrac{\sqrt{23180}}{29} = 5.24$ feet Diameter. Answer.
2. $\dfrac{\sqrt{24000}}{30} = 5.15$ feet Diameter. Answer.
3. $\dfrac{\sqrt{60000}}{24.63} = 9.94$ feet Diameter. Answer.
4. $\dfrac{\sqrt{44000}}{29} = 7.23$ feet Diameter. Answer.
5. $\dfrac{\sqrt{30000}}{30} = 5.77$ feet Diameter. Answer.
6. $(31 \times 5)^2 = 24025$ lbs. Thrust. Answer.
7. $(21 \times 14)^2 = 86436$ lbs. Thrust. Answer.
8. $\dfrac{\sqrt{25000}}{28} = 5.64$ feet Diameter. Answer.
9. $\dfrac{\sqrt{91476}}{21} = 14.3$ feet Diameter. Answer.
10. $\dfrac{\sqrt{188051}}{25} = 17.3$ feet Diameter. Answer.

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